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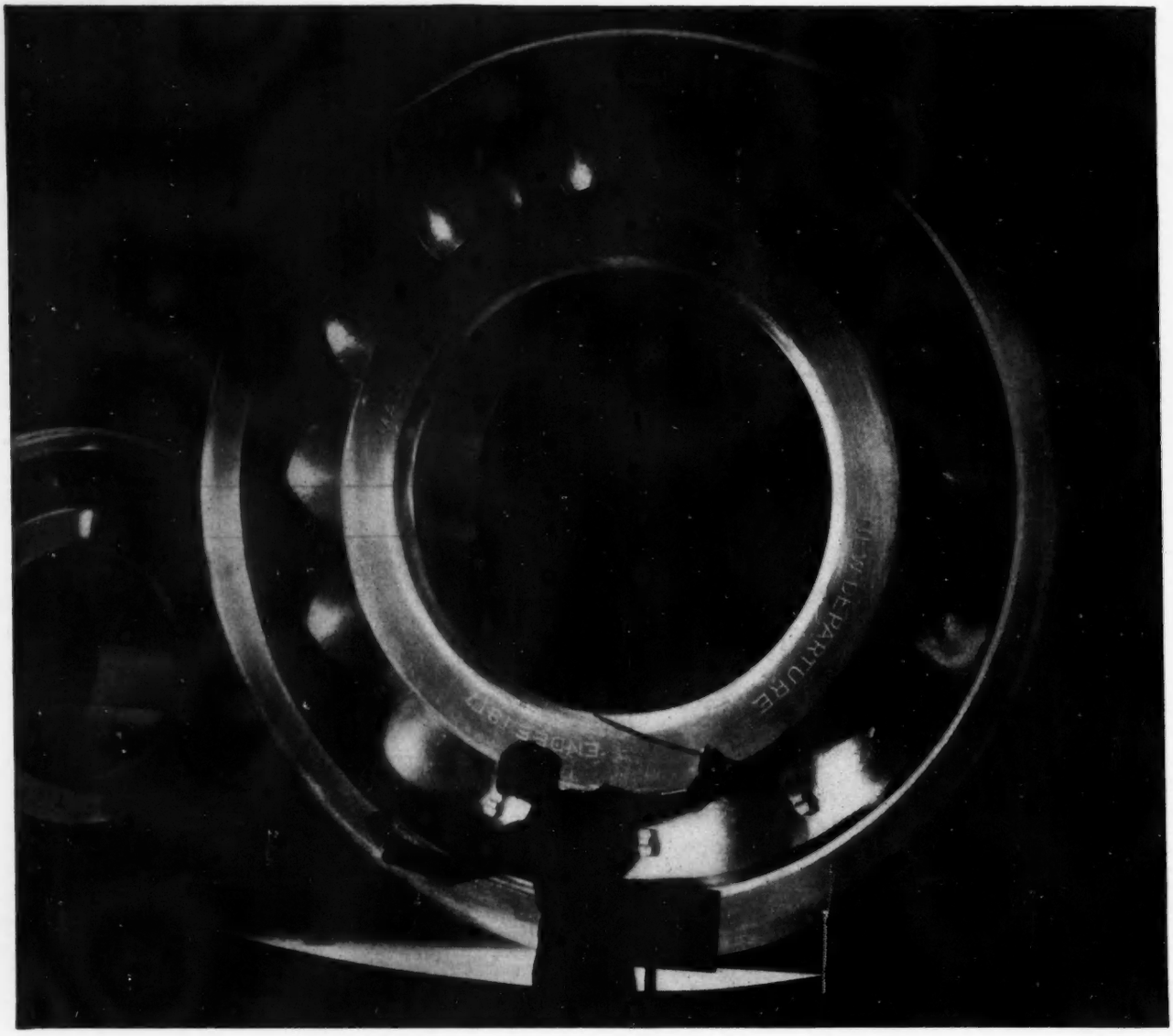

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NEW DEPARTURE BALL BEARINGS

Independent Wheel Suspension

By Georges Broulhiet
Ingénieur des Arts et Manufactures

THE study of composite front axles, called fallaciously "independent wheels," deals with the accelerations due to the inequalities of the road surface and with the resonance of the car caused by the vibrations thus produced. This study cannot be treated separately but must be taken as part of the complete problem of the resonance of the whole car to the road, according to Mr. Broulhiet.

Extensive experiments have confirmed that vertical suspension, road-holding qualities, coachwork resonance and the resonance of the steering gear are all interdependent. Therefore, the problem of road resonance must be dealt with as a whole, if one wishes to grasp the problem in its complexity and have that scientific backing necessary for rapid progress in the mind of the designer.

Mr. Broulhiet outlines, from personal research in this field, four basic conditions to obtain the best results. The geometry and equilibrium conditions of proposed mechanical realizations are analyzed. Comments upon the reception accorded the principles stated, by research organizations and by the public, are made.

Appendix 1 deals with Broulhiet-system shock-absorbers having damping proportional to the speed of the oscillations. Appendix 2 is devoted to statements regarding parallelogram-type independent front wheels. Appendix 3 is a "note" on independent front wheels of the slipper type.

INHERENT to an exaggerated degree in all touring cars of about ten years ago were the following defects, which would not be tolerated nowadays:

The steering wheel vibrated in the hands of the driver;

The car was a source of noise vibrations; the headlamps, the fenders, the glass panes, the coachwork panels, the scuttle-type hood and the windshield were the seat of strong rhythmical vibrations;

The suspensions ruined the road surface, and both the seats and back-rests were in a state of continuous vibration.

But, all these defects have now been greatly reduced, although they still exist to a lesser degree.

The present increased comfort is due, partly, to better roads, but also to the use of low-pressure tires which give a softer contact with the road. It is also due to the patient research of an exclusively experimental order in making the car parts more rigid and using all-steel bodies, more silent mechanisms and an improved system of engine mounting.

The most rebellious of these vibrations—that in the steering wheel, the cause of which has been more thoroughly studied—has been found to disappear by the application of independent wheels when these are judiciously fitted. From this study, of which the "dryness" is at least a certitude, has emerged a general study of the vibrations in cars. The result has been the formulation of a scientifically backed method of progress, which, following the large amount of experimental work done, permits us to proclaim that the *automobile can be driven on modern roads without its experiencing any vibration of whatever kind it be*. This result, which seemed impossible ten years ago, is now practically a certainty.

Faith in this result is derived from the exactitude of the theory, called by scientists the theory of resonance. This theory, which is all-embracing, states that all rhythmical vibrations consist of the vibration of a naturally resonant organ under the influence of an excitation. The car fenders, the glass panes, the doors, the chassis, considered in its bending and torsion movements, are sound-boxes characterised by their musical note; for example, by the number of vibrations per second they produce naturally after receiving a blow. The accidental excitations are the lack of balance of the

[Mr. Broulhiet presented this paper at the S.A.E. International Automotive Engineering Congress, Chicago Sept. 2, 1933.]

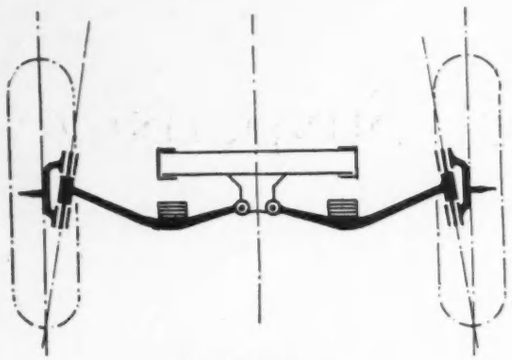


Fig. 1—Split Axle

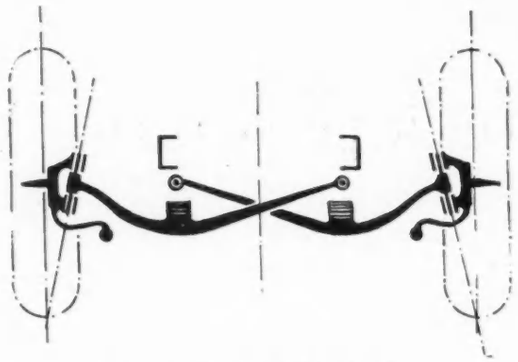


Fig. 2—Two-Legged Split-Axle

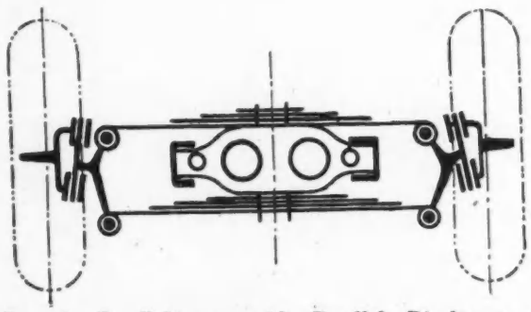


Fig. 3—Parallelogram with Parallel Displacement; Amedée Bollee, Mase

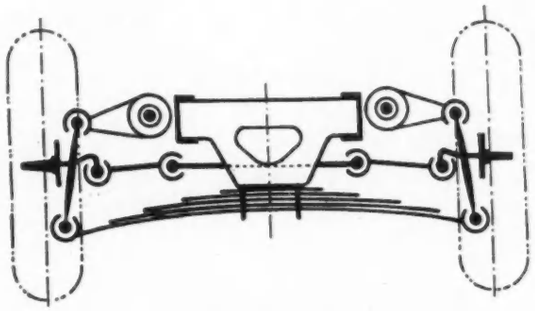
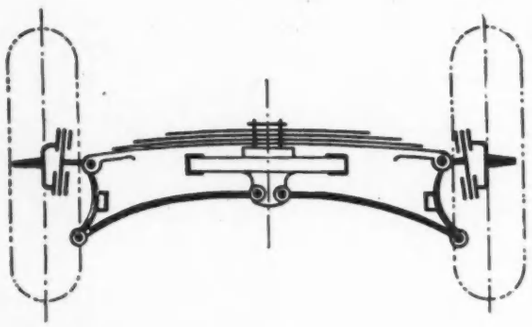


Fig. 4—Constant-Track Parallelogram; Sizaire



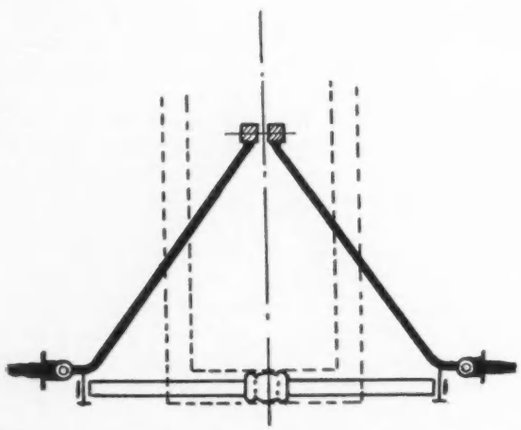
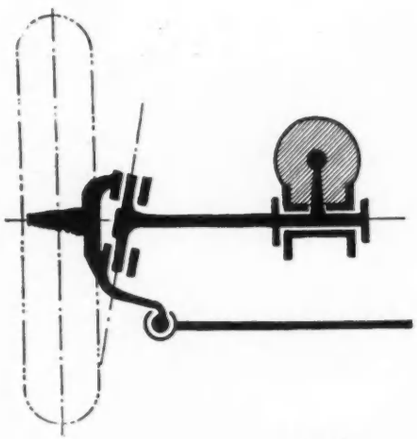


Fig. 5—Parallelogram Based on a Compromise between the Parallel Displacement and Constant-Track Peugeot



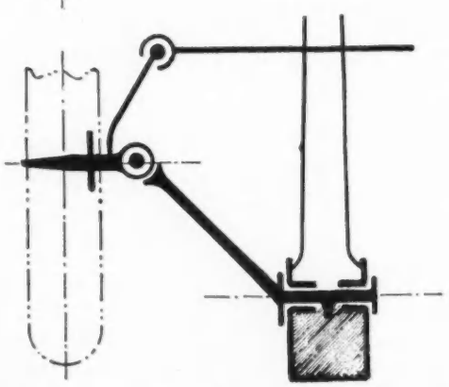


Fig. 6—Elevation and Plan Diagrams of the Léone Laisne Front Axle

engine, the transmission, and the rebound of the wheels on the ground. The periodical excitations are the obstacles met on the road.

On any rotating bodies, such as the front wheels, the gyroscopic couples establish an interdependence; and it is on account of this resonance, which is characteristic in front-wheel shimmy, that the application of the theory began, perhaps because it was the most difficult.

This theory, which is not exclusive in the automobile, is very precise and the experiments which have been carried out on cars have once more confirmed it. Its application requires, however, that those constants necessary for the calculations be determined. These constants are either calculated or else found experimentally in the laboratory. When their value is known with correctness and when their optimum value consistent with manufacturing problems has been arrived at, it is possible to put on the road a car which, from the beginning, will be completely free from vibrations, purely on laboratory preparation.

The interest attached to this method of solution can be immediately realized if it is applied in its entirety and backed by all the necessary means, a condition which will only be completely realized when the industry leaders completely share our convictions.

In this study, which you have kindly asked me to present to you, I have not stated the development already made in France but have tried to give it in descriptive form, of easier comprehension. Before commencing I have thought it better to give a short descriptive foreword, particularly in regard to the terminology used in those solutions of the problem which have already been described, in France, to our scientific institutions.

The problem of cars fitted with independent springing has been dealt with, lately, by Mr. Julien in a paper he read to the Société des Ingénieurs de l'Automobile, of France. Some of the drawings which illustrated this paper are reproduced as Figs. 1 to 9 inclusive. Figs. 1 to 5 represent Transverse Displacement; Figs. 6 and 7, Longitudinal Displacement; and Figs. 8 and 9, the Slipper Type.

The scope of the present paper is twofold: (*a*) to inform our American automobile colleagues of the work we have done in France; and (*b*) to convince the industry leaders of the necessity for equipping their laboratories for the study of resonance.

The Four Conditions of Non-Resonance of the Automobile to the Road.—The comfort of the car, from the point of view of the accelerations, small or large, transmitted to the occupants, is dependent on two problems:

(1) The accelerations due to the engine. This problem deals with the resonance of the car to the vibrations of the engine.

(2) The accelerations due to the road surface. This problem deals with the resonance of the car to the vibrations due to the inequalities of the road surface.

The study of composite front axles, called fallaciously "independent wheels," deals with the second problem, (2).

In both theory and practice, the study of composite front axles cannot be treated separately but must be taken as part of the complete problem of the resonance of the whole car to the road. Experiments have confirmed that vertical suspension, road-holding qualities, coachwork resonance and the resonance of the steering gear, are all interdependent. Therefore, the problem of road resonance must be dealt with as a whole, if one wishes to grasp the problem in its complexity and have that scientific backing necessary for rapid progress in the mind of the designer. My research in this field has led me to formulate the following basic conditions to obtain the best results.

First Condition

Elimination of oscillation of the steering wheels; effect of the *"envirage"* of the tires; necessity for friction in the steering pins; higher efficiency possibilities of steering boxes; and critical speeds, are the subjects included.

The tire *envirage* should be as small as possible. The *envirage* factor determines the steering constants. I have developed this point in a paper[1] I read at the Société des Ingénieurs Civils de France on May 22, 1925. At this conference, I described a characteristic of elastic tires which had not yet been discovered and to which I have given the name of *Envirage*.

Assuming that we have an automobile or a simple form of chariot fitted with four elastic tires in which the steering gear has been rigidly fixed and the front wheels locked so that no movement is possible around the steering pins, let us place the car on a mathematically level plane and then let us push it. The wheel tracks on the ground will be straight lines. Let us once more place the vehicle in its original starting place and let us apply a transverse force to the vehicle that passes through its center of gravity. Again let us push the car in a forward direction. The wheel tracks on the

[1] See *Proceedings of the Société des Ingénieurs Civils de France*, Bulletin for July-August, 1925; see also the *Journal of the Société des Ingénieurs de l'Automobile* (France), June, 1929.

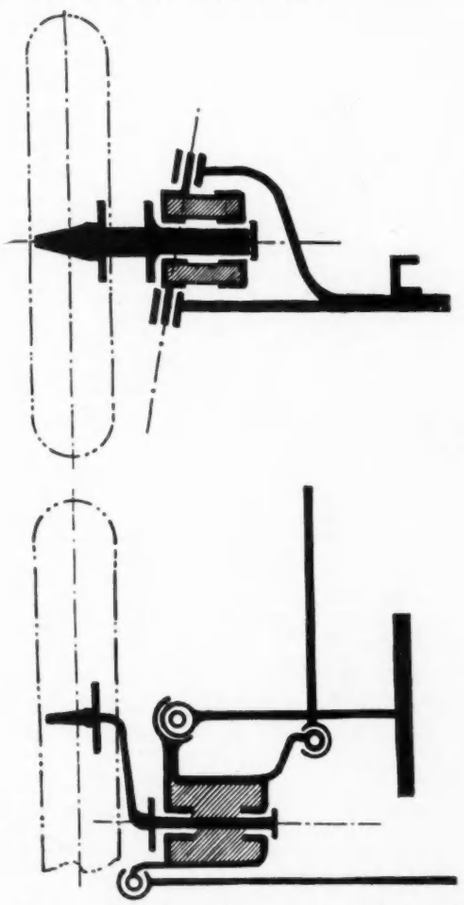


Fig. 7—Elevation and Plan Views of the Dubonnet Front Axle

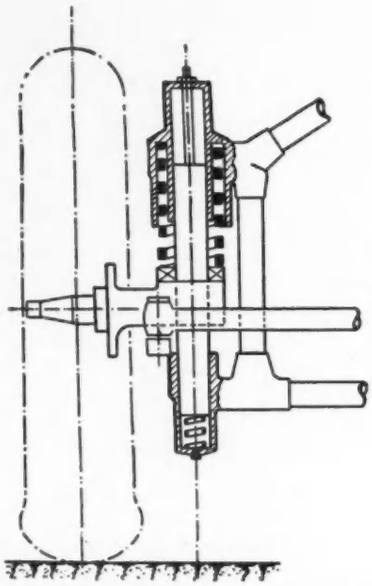


Fig. 8—Elevation Diagram of the Lancia Front Axle

ground will once again be straight lines, but they will be inclined at an angle θ to the preceding ones. The angle θ is a function of the weight of the vehicle and of the transverse applied force.

In the following explanation we will suppose the weight of the vehicle to be constant and this angle proportional to the transverse force. I have named this deviation from a normal trajectory, *Envirage*.

The study of this characteristic is given in the bibliography accompanying this paper, but it can be resumed by saying that the "snaking" or "yawing" movement, or transverse oscillation of a vehicle on its tires, is dependent on the energy borrowed from that caused by the movement of the vehicle itself; this energy has, as an effect, the continuation of the movement. The snaking movement is independent from that of the gyroscopic effect, more commonly known as shimmy. The snaking movement is to be found even in those drives which completely satisfy the conditions necessary for the elimination of the gyroscopic effect.

The elimination, or at least the diminution of the envirage effect, depends on some new quality in the tires. On this point progress would consist in asking tire manufacturers to discuss the problem, as it is possible that this discussion might lead to a new scientific technique of adaptation of tires on cars. At the present moment, tire manufacturers impose their conditions on car manufacturers by purely commercial impositions. This, at least, is true for France.

We must, however, admit that this commercial despotism has given, up to the present, the best results, and I do not think that a scientific competition would have led us, in France, to any greater progress than that already attained. This is due to the excellence of the laboratory work of our tire manufacturers, who have had, up to the present, nothing to learn from car manufacturers in the application of tires. From an historical standpoint, the manufacturer designed his cars to travel, not on the road, but on a set of tires, and the working conditions of these tires are fixed for him by his supplier.

The importance of the problem of the *envirage* and the necessity for progress leading to its diminution can be understood easily by taking the following considerations into account:

It is impossible to center a wheel and its tire dynamically. As a result, the geometrical axis of the wheel does not coincide with its principal axis of inertia, around which it tries to rotate during its natural movement. With every revolution of the wheel, a centrifugal force is exerted, producing a periodical oscillation excitation of the steering train. The frequency of the excitation is equal to the number of revolutions of the wheel; there is, therefore, every possibility of resonance between this excitation and the natural pendular movement of the steering train on its pivots.

I have measured the period of the pendular movement of the steering train on its pivots, and have found it to be equal, approximately, to 0.1 sec. even when the steering link is removed. It is approximately constant for all cars because it depends only on the torsion torque of the loaded tire around a vertical axis passing through the center of its impact on the ground, and on the mass of the wheels and track rod. The quotient of these two constants is of approximately the same order for all cars. There will be, therefore, a maximum of what is called, by French manufacturers, "pseudo-shimmy," at a speed corresponding to a wheel rotation of about 10 revolutions per sec.

The exciting force to resonance increases with the square of the speed; as a result, the curve shown in Fig. 10, which gives the resonance for constant excitation, is altered as shown in Fig. 11.

The realization of this effect can be seen in practice in Fig. 11, and its cause lies in the lack of balance of the wheels.

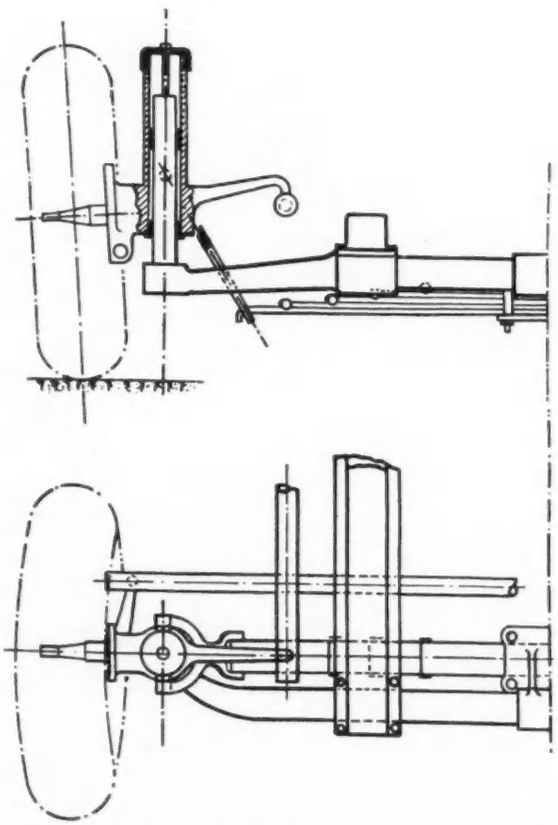


Fig. 9—Elevation and Plan Diagrams of the Broulhiet Type of Front Axle

The reason for its amplification and its continuation lies in the *envirage*. The reason for its diminution and disappearance lies in the passive resistance of the steering gear. It is impossible to get rid of this phenomenon on a car in which the steering pins are completely free.

The passive resistances of the usual mechanism of the steering pins are, naturally, sufficient to damp this effect, even when these pivots are mounted on ball bearings. The inclination of the pivots plays a larger part in the damping than it is usually thought. A few results of some experiments are now given.

The steering torque, measured on the pivots, is, for the two front wheels resting on a cement floor, of the order of 180 lb-ft. The torque was measured after the angle of elastic displacement had been covered and the tire was commencing to slide on the ground. The torque passes from 165 to 195 lb-ft. for the "comfort" to the "supercomfort" Michelin tire mounted on a Citroën C-6-F car.

The constant torque necessary to stop the movement, at resonant speed, is of the order of 15 lb-ft. for all cars, whatsoever be the system by which the wheels are pivoted. This

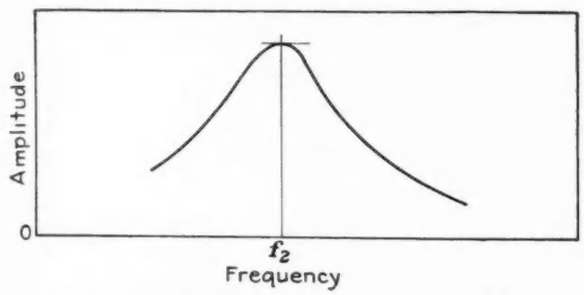


Fig. 10—Resonance Curve Resulting from Constant Excitation

value of the damping couple corresponds in Fig. 11 to the extinction of the amplitude indicated from O to a_2. If the damping torque is diminished, the resonance appears for a frequency f_1, and is a maximum for a frequency f_2; it disappears for a frequency f_3, to reappear again at the higher frequency f_4.

It is possible, therefore, in a new design of independent-wheel suspension, if the mechanism is made with a view of diminishing the usual friction in the steering gear, that vibrations be transmitted to the steering wheel. In this case, it is only necessary once more to renew the friction torque to cause the vibrations to disappear. It can be seen that, after the resonance at frequency f_2 has disappeared, a frequency f_5, which is often out of the range of practical use of the car, appears. Fig. 11 shows that a constant damping is necessary, and experiments have demonstrated and confirmed that even though a damping effect, a function of the speed, does diminish the oscillation movement, it never completely eliminates it.

For instance, on a car having independently sprung wheels designed and made as perfectly as it was possible, a "pseudo-shimmy" was observed at a speed of about 37 m.p.h. that had a maximum at 40 m.p.h., disappearing at 45 m.p.h. and reappearing at 57 m.p.h. On re-establishing the friction couple the vibration at 40 m.p.h. disappeared, to reappear at 66 m.p.h. at a frequency f_5, as shown in Fig. 11.

These experiments are extremely useful to determine the choice of steering box; and a few of the results obtained with a simple classical-type front-axle, as compared with the best

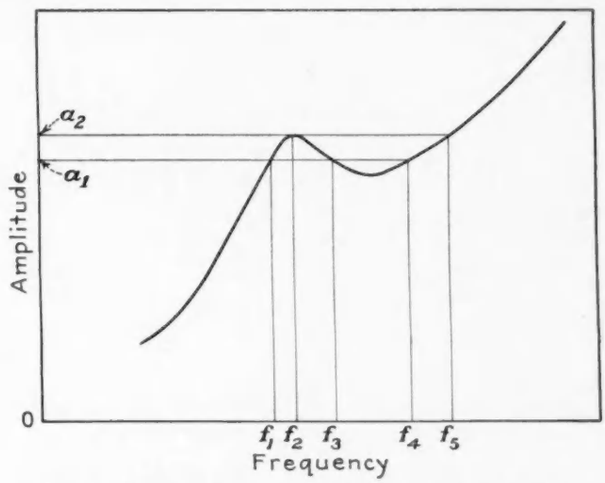


Fig. 11—Resonance Curve Resulting from Excitation That Increases as the Square of the Frequency

type of composite axle with independently sprung wheels, are now given.

Simple Axle.—Pivot resistance, torque of 22 lb-ft.; steering torque at the wheels, 180 lb-ft.; and steering gear with hard spot, the efficiency of the gear beyond the hard spot being 55 per cent.

Composite Axle.—Pivot resistance, torque of 22 lb-ft.; steering torque at the wheels, 180 lb-ft.; and completely free steering everywhere, the overall efficiency being 95 per cent.

With equal steering ratios, the force necessary to rotate the steering wheel of an independently sprung wheel system can be brought to $55/95 = 58$ per cent of the effort necessary on the steering wheel of the simple-axle system.

Experiment has shown that the steering ratio can be increased, there being no change in the driving impression, when passing from a ratio of 15:1 on an ordinary axle to the ratio 18:1 on a composite axle. This is probably due to the absence of lag on the steering wheel, which, in the composite axle, remains at the center of the play. In my opinion, with a good composite axle, the following constants can be adopted:

Total torque on the steering pivot with "supercomfort" tires, 200 lb-ft.

Angular ratio between the steering wheel and the wheels, 13:1.

Steering-box efficiency, 95 per cent.

Reduction of parking effort, $(13/15) \times 58$ per cent $= 50$ per cent.

I should like to point out that both the Sizaire and the Adler axles are fitted with steering boxes of the rack-and-pinion type, of very high efficiency; but both designers had to resort to a diminished angular ratio between the steering wheel and the wheels—between 8:1 and 11:1—as, for a normal ratio and with the necessary length of steering levers, the steering-box pinion would have been so small as to render its manufacture impossible. Notwithstanding the low steering-ratio, the parking effort necessary is still low if compared with an ordinary steering box, but as regards the driving of the car, the steering is "too direct," as it is called in France. It is very debatable whether this result, which might be advantageous in a racing car, is also advantageous for a touring vehicle, and the point is very much under discussion. It is therefore necessary to create, for independently sprung front wheels, a steering box of very high efficiency, having at the

same time a sufficiently low ratio. For this reason, steering boxes of the screw-and-nut type are preferable to those of the worm-gear type. A steering box of the spur-gear type does not seem possible except by using a double train of gears.

In France, although I stated the problem very exactly to those manufacturers who utilized my ideas, I have not as yet been able to obtain a design of steering box answering to the very precise requirements of independently sprung wheels; that is, a box having a 95-per cent efficiency, with a ratio of 13 : 1.

The steering defects, amplified by *envirage,* can therefore be eliminated easily by compensating them by the friction in the pivots; and this can be done with great benefit in the

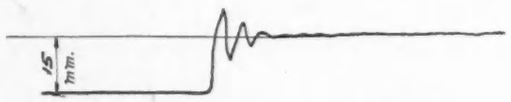


Fig. 12—Vertical Movement of the Wheel on the Tire
Tire: 140 x 60 Michelin; Inflation Pressure: 1 kg. per cm.² Load; Ordinary Front Spring: 340 kg. A 70-kg. weight is attached to the wheel, representing one-half of the weight of the complete front axle. Deflection of the Tire: 15 mm. Period: 0.1 sec.

case of independently sprung wheels, as the efficiency of the steering box can be considerably increased. There is a second disadvantageous effect which is a function of the *envirage;* that is, the screaming of the tire on road curves. On the contrary, the adherence is not diminished; however, I leave this musical consideration to musicians.

Second Condition

Elimination of shimmy; independently sprung wheels require a rigid reference plane, and the theory of bouncing shimmy are the subjects included.

It is essential that the energy due to the *envirage* picked up from the ground be not transferred by the gyrostatic couples in the bouncing movement of the wheels on the ground. This is effected by obliging the wheels to remain parallel to a fixed plane while moving against the suspension springs. This means, in other words, the substitution of the so-called "independently" sprung wheels to the ordinary classical axle.

This problem has already been dealt with by me, from a theoretical standpoint, in the paper[2] I read May 22, 1925, to the Société des Ingénieurs Civils de France, referred to previously. I shall here repeat the explanation of this phenomenon.

The pendular rebounding movement of the front wheels, and the snaking movement around the steering pins, have each, taken individually, a natural period of oscillation of their own. These motions are modified by their becoming interdependent as soon as the wheels rotate. This interdependence is caused by the presence of the gyrostatic couples, and, when the gyrostatic torque grows, these two movements draw away progressively from their individual periods, but without the possibility that their periods will ever become one common period because the gyrostatic couples cause them to draw farther apart instead of making them approach one another. It is therefore possible that two motions can be the result; a snaking motion around the steering pins, and a rebounding motion.

This snaking motion is similar to the "pseudo-shimmy" which I have described in a preceding paragraph. This snaking movement has its period near to that of the natural period of oscillation around the steering pins. It is stimulated directly by the *envirage,* and the modification of its period of oscillation is influenced by the value of the gyrostatic couple. It remains in the form of pseudo-shimmy if the gyrostatic couple disappears.

The bouncing of the wheels can only be maintained by the gyrostatic torques, deriving their energy from the first movement of the said snaking movement, which, in turn, is stimulated by the *envirage* of the tires. With the complete elimination of *envirage,* no continued movement could persist; shocks on the steering wheel would die out after one or two oscillations, though they would not be completely eliminated.

The composite axle, with parallel displacement of the wheels, has the following effects:

(1) It can prevent the creation of blows or shocks on the steering wheel

(2) Suppression of all derived energy in the rebounding of the wheels; that is, opposition to all vertical shimmy

(3) The mechanism called "independently" sprung wheels cannot prevent the oscillating movement or pseudo-shimmy, described in preceding paragraphs.

Summing up, the elimination of *envirage* would have for effect the prevention of all permanence in the oscillating movements of the steering gear, but would not eliminate shocks on the steering wheel.

The elimination of the gyrostatic couples has for effects the prevention of shocks on the steering wheel and the suppression of the vertical rhythmic movement of the front wheels. This has no effect on the pseudo-shimmy, which can only be eliminated by the frictional resistance in the pivot pins.

It is useful to note at this point that, to suppress the gyroscopic couples, it is not enough that the front wheels be parallel with each other in their displacements; they must also

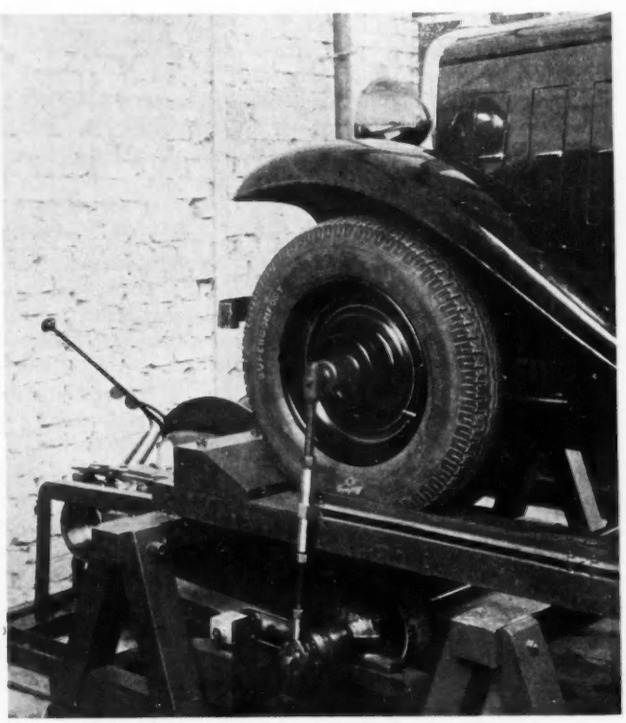


Fig. 13—Vibrating Machine Attached by a Connecting Rod to the Journal of a Front Wheel

[2] See *Proceedings of the Société des Ingénieurs Civils de France,* Bulletin for July-August, 1925.

remain parallel to a rigid reference plane. The details of the calculations regarding these gyroscopic couples can be found in my Société des Ingénieurs Civils de France paper[3].

A *qualitative* knowledge of this phenomenon, however, is sufficient in itself to give the direction for a new design of a car. A scientific study of all the conditions of the problem, taken *quantitatively,* would only lead to a diminution of the defects in the classical type of axle, and even then it would be essential to include in the calculations all the elastic coefficients of the steering train, a condition which seems rather impracticable. On the other hand, the complete elimination of the gyrostatic couples gives the result directly, by eliminating the fundamental causes. Therefore, any calculation of the effects is eliminated with the elimination of the cause itself, and the problem becomes relatively easy even for an unskilled designer.

Let us now suppose a case in which the parallelism of the wheels and the damping effect in the pivots have eliminated all resonances in the car which were due to the steering train. The remaining resonances are of two kinds: (*a*) the movement of the whole of the vehicle, which is supposedly rigid, and (*b*) the resonances of the individual components of the vehicle in relation to the coordinates of their positions.

The resonances of the individual elements composing the vehicle are in strict dependence with the conditions of parallelism of the plane of the front wheels, and, in practice, these two problems are treated together in the one and only problem of the rigidity of the whole.

Third Condition

The problem of rigidity; resonance of the chassis; effects on the steering gear; effects on the comfort of the suspension syestem; as absolute rigidity does not exist, what coefficient can be considered sufficient; and experiments, are discussed.

The rigidity of a chassis has two effects. First, an absolute rigidity would be necessary from the point of view of suppression of all gyrostatic actions of the wheels. For this reason it is necessary that the front wheels should move parallel to one another, which is the same as saying that there exists a rigid immovable plane in space to which they can be referred.

The greatest immobility of this plane can only be obtained by concentrating the heaviest portions of the car on a supporting member, which must have the maximum of rigidity possible. With independent front-wheel springing, the nondeformation of the front of the chassis acquires an importance which was not required in the ordinary classical type of axle. In fact, in the latter, the motion of the axle is separated from the chassis by the springs, and the movement of the axle is only transmitted to the chassis in a very reduced form. The chassis, therefore, practically has no reaction on the axle.

In the independently sprung wheels, the oblique movement of the axle which we are trying to suppress is, in effect, the movement of the front of the chassis itself, on which the wheels are fixed. Therefore, the front portion of the chassis must have a very large torsional resistance against a torque, due to a blow, around the longitudinal axis of the car. An analytical expression for this quality is complex, as it takes into account the masses successively attached to the chassis and also the elasticity of those parts of the chassis placed between these masses and the wheel fixture.

[3] See *Proceedings of the Société des Ingénieurs Civils de France,* Bulletin for July-August, 1925.
[4] See *Omnia* (France), February, 1933.

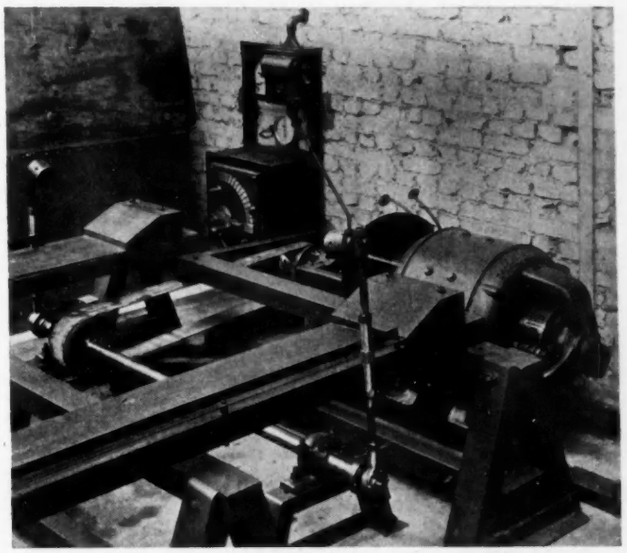


Fig. 14—Details of the Vibrating-Machine Mechanism

It is evident that under a torsional effort the chassis will commence by deforming itself and, under a continuation of the torque, the heavy parts of the mechanism will rotate faster as they are lighter around the longitudinal axis of the car. The effect of changing the plane of the wheels is caused by two rotations added to one another; the rotation due to the deformation of the chassis, and the rotation of the whole.

The two conditions necessary that this effect be the smallest possible are a chassis-deformation factor of the lowest possible order and a concentration of the masses as near as possible to the front wheels. To determine the rigidity factor, I have evolved a method which was described in detail[4] by my assistant, Mr. Hertz.

The second quality is obtained by an extremely rigid fixation of the front part of the engine and of the front part of the body. This second quality is added to those that a chassis should generally possess to give the minimum of resonance in the bodywork; and this is the second effect of the rigidity of the chassis. The rigidity of the chassis has a great influence on the rigidity of the entire bodywork and its accessories, and has a very marked action on the resonance of these parts.

It is impracticable to calculate, *a priori,* the resonance of the various components of a car on account of the considerable amount of data necessary, which, for the greater part, would have to be replaced by hypothesis.

The best method of arriving at a satisfactory solution is by the means of trial and error. By imagining, building, trying and retouching, a solution will be reached in the quickest way, provided that the designer be inspired by scientific considerations indicating which is the quickest and safest means to the solution of the problem. Here are given the elements, calculated and measured, of this practical problem, and the guiding considerations for the study of the resonance due to riding on the road.

The excitations at the road surface are of two types; (a) those due to obstacles and (b), rhythmic rebound of the tires.

The excitations due to obstacles are purely isolated blows; they become periodical if the defects of the road surface become themselves periodical. The interaction of the obstacle and the suspension may cause the obstacle to become peri-

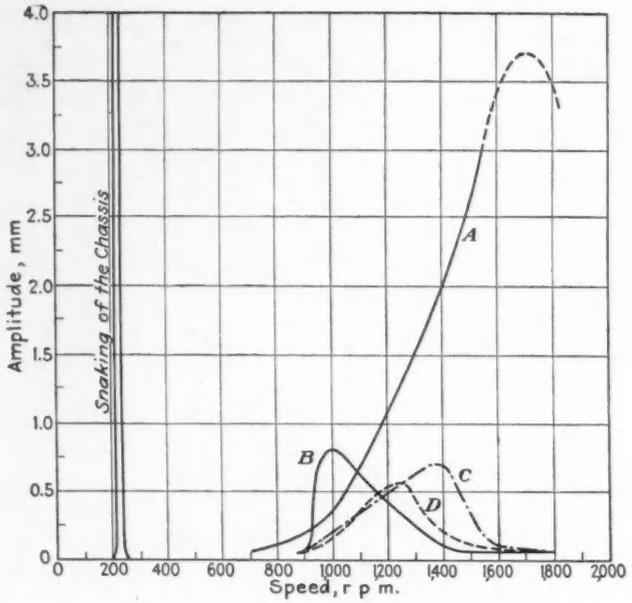


Fig. 15—Maximum Amplitude of the Vibrations at Various Points on the Body; Production Springs
Curve *A*, Fenders; Curve *B*, Suitcase (*Malle*) without Fenders; Curve *C*, Hood (*Auvent*) without Fenders; Curve *D*, Top (*Pavillon*) without Fenders.

odical, by causing wear on the road surface. A rather curious, and nowadays exceptional fact, is that a road used by trucks shakes a car much more than does a road used by horse traffic. I published a study on this subject[5] under the title *Comment l'Automobile Use la Route* (Wear on Road Surface Caused by the Automobile).

The rhythmic rebound of the tire has a constant period, whatever be the speed of the car. This period is easily calculated if the weight of the wheel and of the part of the axle connected to it are known. It is also necessary to know the coefficient of elasticity of the tire. By calculations, it is found that this period varies between one-eighth and one-twelfth of a second.

A test machine has given us the diagram shown in Fig. 12, which demonstrates this period of oscillation of 0.1 sec., agreeing with the calculated value. It is apparent, therefore, that a car going along a road will receive a certain number of isolated blows corresponding to the obstacles met; but, above all, it will receive a continuous excitation due to the rebound of the wheels on the ground every 0.1 sec.

From this result was born the idea of determining, in the laboratory, the resonance due to riding on the road, by exciting the axles by means of a periodical movement applied every 0.1 sec. There are two types of machines:

(1) A *qualitative* machine, to discover the resonances and to study, practically, by laboratory tests, the best means to diminish or to cause the vibrations to disappear.

(2) A *quantitative* machine, to study the resonance of those parts of the car at the vibrations which will have been found to be impossible to avoid, after the qualitative investigation, and principally that due to the rebound of the wheels.

The first of these two machines was built in the laboratory of the Citroën Works. It is illustrated in Figs. 13 and 14.

A shaft placed under the car and driven by an electric motor causes a series of vertical oscillations to be impressed on the front wheels. The total amplitude of the oscillations is 4 mm., with a difference of phase of 180 deg. between the left wheel and the right. To this end the shaft carries two eccentrics at its extremities, having an angular relative displacement of 180 deg. and an eccentricity of 2 mm. These two eccentrics are connected, one to the center of the right wheel and the other to the center of the left wheel, by means of links with spherical joints. The shaft speed can vary between 100 and 2200 r.p.m.

This machine has given the beautiful diagrams in Figs. 15 to 18, which agree entirely with the classical theory of a maximum of resonance, which I exposed in a conference at the Société des Ingénieurs de l'Automobile, of France, in 1930, regarding the study of gear noise[6].

Curiously enough, this machine showed us, with extraordinary precision, a phenomenon which had been practically forgotten in France. A very individual form of resonance to the road manifested itself in a very abrupt snaking of the car on its tires, producing instantaneous changes of direction that were capable of sending the car into the ditch on the side of the road before the driver had time to collect his wits. The machine showed, in the laboratory, that this movement or, rather, this resonance, takes place in a very narrow range and at a low number of revolutions per minute of the testing machine, corresponding to a low frequency; therefore, under conditions reproduced with great difficulty on the road at high speed. The testing machine showed that, when this movement takes place, it does so with great intensity. It is the equivalent of the *coup de tabac,* or blow, which happens unexpectedly on large ocean liners and which takes place sometimes after very long intervals; often, not for years.

This first qualitative machine, built entirely on theoretical principles, allows a detailed examination of the whole range of resonances. Experiments have shown that these resonances can all be cured, in general, by increasing the rigidity of the components of the car. The two resonances eliminated with the greatest difficulty, when they appear, are the vibrations of the fenders and the scuttle-type hood. I have found that, when these two are present, the fender supports and the shroud panel have to be redesigned because adding strengthening members has only a very small effect.

For the fender vibrations, the method of least rigidity—lengthening of period—leads more often to complete immobility. Unless aerodynamics gives us a new type of thick fender, I believe that the articulated-support types such as Chevrolet, Studebaker and my own, are the best.

For the scuttle-type hood, those manufacturers who have followed the school of obtaining the maximum of rigidity through the molding of this hood have had no difficulty, provided that the connection between the hood and the chassis is made perfectly.

The desired results for the car on the testing machine are obtained by comparing the resonances of the cars among each other and taking as a model the minimum found. The amplitudes are checked by mirrors, the periods with tuning forks, and the excitation period is obtained directly by obtaining the number of revolutions per minute of the eccentric on the machine. The tests should be carried out on a completely equipped car, and with varying numbers of passengers. By this means the interdependence of the various parts of the car can be found. One of the most curious to notice was the effect on the steering column of a scuttle-type hood lacking in rigidity.

[5] See *Revue de l'Ingénieur* (France), April, 1924.
[6] See *Proceedings of the Société des Ingénieurs de l'Automobile* (France), May, 1930.

The second testing machine is used to test the resistance to resonance. It is a quantitative machine of the well-known shaker type driven at constant speed. The following machine constants were chosen and kept constant throughout the tests: Amplitude of movement, 1 in.; speed of rotation, 600 r.p.m.

Breakages occurring during a test and caused by this machine take place very quickly, usually within 20 min., and the results are always comparable between tests. This machine has been useful principally in the determination of weak points of my box-frame rigid chassis, and this has been a great help in perfecting welding processes in the workshop.

The laws of rigidity for a chassis must be applied to the chassis and body taken as a whole, and can be summed up under the following rules:

(1) Maximum torsional rigidity. The inertia of the car must be principally localized near the front axle.

(2) The separate rigidity of each of the components of the car must be determined in the measure demanded by the cessation of resonance. A diminution of the rigidity is admissible only in the case of the attachment of the front fender to the chassis, because, in this case only, it is easier to go below the resonance period than above it.

Fourth Condition

Vertical Suspension.—Springs and shock-absorbers. The oscillations are induced chiefly by the constant factor of damping. Damping of the body oscillation on its springs and damping of the axle oscillation, or double damping. Damping of the return of the springs depends on the car constant, and damping on the in-stroke depends on the road and on the tire.

The disappearance of all resonance on the steering gear makes driving comfortable and safe.

The disappearance of resonance in the body-chassis unit makes for a silent car.

It is necessary now to establish the vertical suspension by springs and shock absorbers, without any other limitation beyond the conditions of equilibrium.

In 1911 I studied and determined a psychological scale of vertical vibratory accelerations, and published an article[7] on this subject.

We know, now, that for passenger cars, static spring deflections under load ranging from 5 to 8 in. can be adopted in the front axle, while deflections from 8 to 11 in. can be used on the rear. The value for the static deflection under load is limited by the following considerations: Equilibrium on road curves, bumping play and loss of level of the car under variable load. A larger value for the static deflection under load could be adopted only by using a spring system fitted with a vertical adjustment.

In the first place, the usual type of spring is an accumulator of elastic energy, and the law of effort is a linear function of the displacement. The characteristic of the spring is its weight; if it is built up with fancy spacing, by uselessly strengthening it in those places where it is believed that rupture will occur, and where it never does, it is capable of accumulating from 40 to 60 lb-ft. per lb. of its weight.

The design of springs built up on strictly arithmetical spacing has given, with ease, an accumulation of energy amounting to 72 lb-ft. per lb. of ordinary spring steel.

If deflections, under load, of 7 in. for the front axle and 10 in. for the rear are provided for, the weight of the spring is found to be 2.2 per cent of the weight of the loaded vehicle; whereas, in a clumsy determination of the spring weight, it can even reach the figure of 4 per cent of that of the loaded vehicle.

Torsion springs are much lighter for the same quality of steel. Theoretically, they weigh only five-twelfths of the weight of leaf springs, which would correspond to a weight of only 1 per cent of the loaded vehicle.

On the front independent springing of Lancia and Porsche cars, torsion springs are used. In the first, they are of the ordinary helical type; whereas, in the second, a plain rectilinear bar is used.

In the Porsche car the lightness of the spring has been still further increased by the use of steels having high elastic limit; and, as the weight of the springs is inversely proportional to the square of the maximum stress of the steel, it has been possible to reach, for the spring of a front wheel, a weight of only 800 grams of steel for a car weighing 1100 kg. when empty. We would like to point out, however, that, with the dimensions used in this design, an ordinary manganese-silicon steel would not have stood up to the stress imposed. To lighten the design still more, compressed air would have to be resorted to.

Should the torsion-spring school be followed? Yes, but only under the condition that its installation be easy, as is the case in the longitudinal parallelogram type, with very short levers. If the levers are long, the necessity for strengthening them to withstand the imposed strain increases the weight and ruins the economy of the system. In this case the leaf spring is simpler and less handicapped by its weight.

The hairpin type of spring as tried by the Germans on their railway carriages and abandoned a long time ago, and the helical type of spring, are not often used in car suspension, notwithstanding the economy of steel. The problem centers around the choice of installation system of the torsion springs; it, therefore, should be employed every time that a proposed suspension system incorporates a rotating axis with short levers.

The springs should operate without any friction whatsoever, as is the case in torsion springs. The usual types of leaf springs, having a damping effect due to the friction between the leaves, are bound to disappear.

By calculation, it is known that the oscillation of a suspended mass is caused by a blow, $\int Adt = mV$, which determines the initial speed of the suspended body after the impression of the force; for example, an obstacle met by the

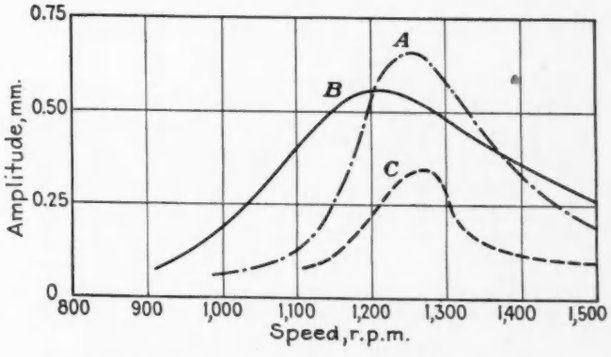


Fig. 16—Maximum Amplitude of the Vibrations of the Hood *(Auvent)* with Fenders

Curve *A*, Broulhiet frictionless springs having a flexibility of 13 mm. per 100 kg.; Curve *B*, ordinary production springs; Curve *C*, Broulhiet frictionless springs having a flexibility of 22 mm. per 100 kg.

[7] See *Revue de l'Industrie Minérale* (France), Sept. 1, 1921.

wheel of the vehicle. This creates an amount of energy $mV^2/2$ which will have to be destroyed by the shock absorber.

These are the only theoretical data necessary to resolve the problem of the suspension. We must, therefore, reduce $\int A\,dt$ to reduce V; now A is a supplementary force on the spring when the wheel passes over an obstacle.

Let us now imagine a difference of level of 10 mm. met by a wheel equipped with a spring having a 200-mm. static-deflection under load. The variation in the spring force will consist of an elastic force equal to one-twentieth of the load plus a force due to friction. If the friction coefficient between the spring leaves is equal to 5 per cent, the reaction on the chassis frame is doubled. Friction must therefore be reduced to zero to be able to reduce to zero the reaction on the chassis frame for the smallest displacements of the wheel axles.

Here is a practical experiment which fully confirms the foregoing rule. A car fitted with springs designed by me, having rollers at the extremities of the leaves, and, therefore, with a very greatly reduced friction force, and in which the shock absorbers were disconnected, can be driven down the Avenue Foch in Paris without giving rise to that rocking motion so well known to Parisian testers. As soon as the rollers are removed from the extremities of the leaves, however, and the usual friction damping effect is re-established, the undulations immediately reappear. This is a very definite case where friction damping produces oscillations. Therefore, the first condition in a spring suspension system is the necessity for elimination of the constant damping factor of the suspension, to avoid the induction of oscillations due to spring compression over obstacles on the road.

I have carried out a complete set of tests on this problem,

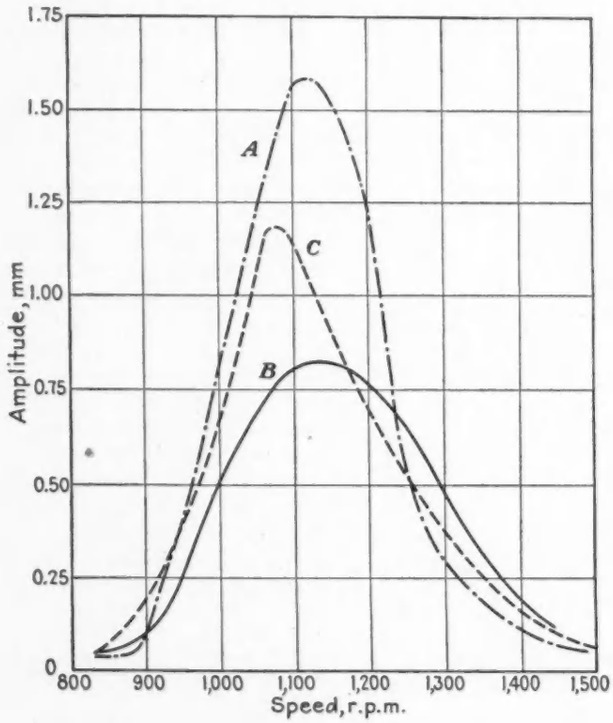


Fig. 17—Maximum Amplitude of the Vibrations of the Suitcase (*Malle*) With Fenders

Curve *A*, Broulhiet frictionless springs having a flexibility of 13 mm. per 100 kg.; Curve *B*, ordinary production springs; Curve *C*, Broulhiet frictionless springs having a flexibility of 22 mm. per 100 kg.

so as to settle once and for all the question of suspension with self-damping springs. Friction between the leaves always gives rise to a galloping effect with insufficient rebound damping of the wheels on bad roads. Besides this, friction between the leaves of a spring never remains constant. I have found that the friction of a spring when new increases by 40 per cent before settling down, which occurs only after about 400 miles. If the spring is now taken to pieces and re-assembled, without greasing the leaves, the spring will again be at its starting point, once more to settle down after a few hundred miles, reacquiring the same value for the friction coefficient as it had the first time.

In another test on the damping of spring oscillations, I have tried to obtain a damping effect proportional to the speed of the oscillation. I reached this result by interposing, between the leaves of the spring—fitted with rollers at the extremities of the leaves—a composition that is called "chatterton," made up of the following substances suitably heated: Gutta percha, 40 per cent; tar *(brai de pétrolé)*, 30 per cent; and resin, 30 per cent.

I have shown in Fig. 19 curves for such a spring, taken on an expansion machine fitted with a magnetic release device. The diagrams given are, first, 65 hr. after the application of the foregoing composition; then, after 100 oscillations; then, after 1000 oscillations, and, finally, after 6000 oscillations. It can be seen that the spring is at first too damped above the aperiodical critical damping point, but settles down rapidly after the first 100 vibrations.

In Fig. 19 I have shown also the curve for the same spring, before the application of the foregoing mixture, both with rollers at the extremities of the leaves and after the rollers have been removed.

The self-damping type of spring with the composition between the leaves, which was perfect on the test machine, became absolutely impossible during the road tests on a car. The damping on spring compression was too strong and induced a perpetual form of gallop, by short immediately damped displacements.

All my experiments confirm the theory that damping on the in-stroke of a spring must be small in intensity, without friction constant. Research in the field of self-damping springs is therefore wrong. After studying this problem for a year at Peugeot's Works, I have abandoned it to take up the study of hydraulic damping.

In the classical differential equation of a pendular motion:

$$M\frac{d^2x}{dt}+f\frac{dx}{dt}+Cx=0 \qquad (1)$$

M is the mass and C is the bending constant or rating for the spring. The terms $M\frac{d^2x}{dt}$ and Cx represent the reversible forces, an inertia force and an elastic force. The term $f\frac{dx}{dt}$ is an energy-destroying force, a damping force introduced into the classical equation in the shape of a force proportional to the speed, with the object of rendering easy both the integration of the equation and the study of the movement.

I thought it might be useful to try to build a shock absorber answering to this analytical concept and to apply it to the damping of the oscillations of the springs of a car. By resolving classical differential equation (1), it is known that the aperiodical critical damping is obtained from

$$f^2=4MC \qquad (2)$$

where f is a coefficient which is constant by definition; that is,

$$f=\frac{\text{Damping Force in Kilogrammes}}{\text{Speed at the Damper Joint, in Meters per Second}}$$

where C is the bending constant of the spring; that is,

$$C=\frac{\text{Force of the Spring in Kilogrammes}}{\text{Deflection of the Spring in Meters}}$$

and where M is the mass solidly attached to the spring; that is,

$$M=\frac{\text{Weight in Kilogrammes of the Load}}{\text{Acceleration Due to Gravity, or 9.81}}$$

In a car suspension, there are two movements which have to be damped; the rebound of the wheels and axle on the ground and the rebound of the car suspended on the axle. The damper, being attached between the axle and the chassis, damps both movements. To enable the damping to be aperiodical for these two movements, it is necessary that the constants f for each of these individual movements, be equal.

Rebound of the Wheels on the Ground—The rating of the elastic system for the pendular motion is the sum of the two constants C_1, for the deflection of the tire, and C_2 for the deflection of the actual spring. The mass subjected to the oscillation is the non-suspended mass m, formed by the wheel and a part of the axle, and

$$f_1=2\sqrt{[m(C_1+C_2)]} \qquad (3)$$

Rebound of the Car Suspended on the Axle.—The rating constant of the elastic system for the pendular movement is only that of the spring. The mass subjected to the oscillation is the mass M, the suspended mass, and

$$f_2=2\sqrt{MC_2} \qquad (4)$$

It is necessary that $f_1 = f_2$ to allow the damping effect to be the same for both the movements; that is,

$$MC_2=m(C_1+C_2) \qquad (5)$$

These data depend only on the design of car, and are independent of the shock absorbers. They establish the best relationship that should exist between the masses of the car parts, the flexibility of the springs and the flexibility of the tire. In the average car, this condition is found to be generally quite satisfactorily followed.

In the Peugeot 201 car, $f_1 = 500$ and $f_2 = 700$; these two figures are sufficiently near. The same calculation has given 700 and 900 for the Citroën C-6-G car.

The design of my shock absorbers, as in that of all good hydraulic shock absorbers, is based on the following principles:

(1) Damping must be independent of the temperature.

(2) The in-stroke speed of the tire due to the road is dependent on these two factors, tire and road, and must be determined experimentally.

(3) The speed on the return stroke of the wheel on the ground depends only on the pendular constants of the car; that is, on the masses and rating of the springs. A damping proportional to the speed of the oscillation of the return is applied, giving an aperiodical damping effect.

(4) For the in-stroke, a damping proportional to the speed is also applied. This is determined so as to give it the same maximum value for damping force as on the return stroke.

(5) For the in-stroke, the damping coefficient is about one-fifth of the damping coefficient for the return stroke, as calculated theoretically.

(6) The force at the joint of the damper on the in-stroke must have some means of limitation on account of the unforeseen obstacles which will be met on the road. The damping on the return stroke requires no limitation, as the maximum force is dependent on the pendular constants of the car.

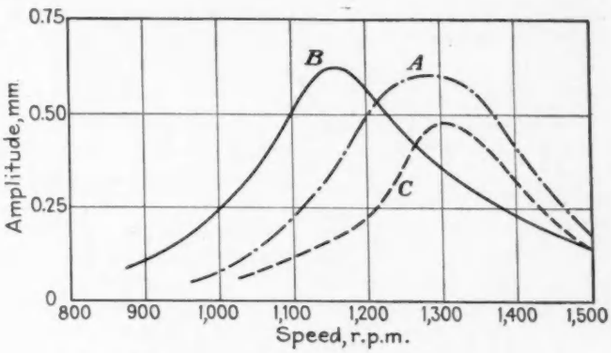


Fig. 18—Maximum Amplitude of the Vibrations of the Top (*Pavillon*) with Fenders

Curve *A*, Broulhiet frictionless springs having a flexibility of 13 mm. per 100 kg.; Curve *B*, ordinary production springs; Curve *C*, Broulhiet frictionless springs having a flexibility of 22 mm. per 100 kg.

I have given a very simple solution for the mechanical realization of a damping effect which is proportional to the displacement speeds and at the same time independent of temperature variations. Some indications on this subject are given in an Appendix to this paper.

I have also given in an Appendix some theoretical calculations executed by my assistant, Mr. Hertz, showing that the effort of the damping varies very much less rapidly than the damping coefficient, which explains that, once the adjustment has been fixed, it can vary plus or minus between 25 per cent for only a variation, plus or minus, of 10 per cent in the force at the joint of the shock absorber.

In the mechanical realization of a composite axle, or independent springing parallelogram type, I am opposed on principle to the use of the spring as a reaction lever. The increased size of the spring's leaves gives an additional weight which is of the same order as a reaction lever, and the displacement diagram is less certain. Only reasons of economy can justify the use of a spring as a reaction lever, and practice will give the final judgment.

Examination of Proposed Mechanical Realizations—Geometry and Equilibrium Conditions.—The plain single longitudinal lever, with a joint in front of the wheel axle and the two-legged split-axle or plain split-axle for the front steering-train are doomed to disappear. The first for two reasons; (*a*) the variations of the caster, which renders steering adjustment inaccurate, and (*b*) lack of longitudinal equilibrium in the braking.

The split two-legged-axle types are only semi-solutions. They are tempting, if the modification of an already existing type of car is being considered. It is in this way that the Peugeot company entered the field of independent-springing tests. Owing to its multiple imperfections these types were rejected as a front steering-train, as they are too far from theoretical requirements. The split axles are less objectionable when used on the rear train, and they have a certain amount of popularity in Central Europe. Its defects, when used on the rear train, are limited to the wear on the tires, and on the necessity for allowing a certain amount of clearance for the lateral displacements of the wheels, thus reducing the seating capacity and the length of the rear cushion. I am therefore only mentioning these two solutions as an "in memoriam."

As a general rule, solutions not respecting conditions of longitudinal equilibrium for braking, and those systems in-

creasing the aptitude of the car to overturning on road curves, should be immediately discarded. The conditions of longitudinal equilibrium depend on a very exact theory[8] which I formulated for the first time in 1921.

The errors made in the lengths of the torque rod and brake-reaction levers give rise to trouble which is accentuated in proportion as the spring flexibility increases. This is exactly the case for the front axle having independently sprung wheels, which can accommodate springs having a flexibility three or four times that of a spring mounted on the classical type of axle, and all this to the benefit of the suspension.

Some fallacious scientists say, when demonstrating how the brakes crush the car to the ground, "What remarkably powerful braking!" This "crushing" effect of the car on the front or on the rear—for both conditions exist—throws the passengers in the car forward and reduces uselessly the displacements of the springs. The more a car remains horizontal while braking, the more pleasant it is. The lack of longitudinal equilibrium, on the other hand, transforms the torque on the driving wheels into an excitation for the galloping oscillation of the car.

Correct Equilibrium

It must be remembered that on a car having the rear wheels as drivers, and with brakes only on the rear, the correct equilibrium of the rear axle requires equal proportion, on the one hand, between the wheel radius and the height of the center of gravity, and, on the other hand, between the torque-rod length on the rear axle and the distance between the two axles. In cars in which all torques and thrusts are taken by the rear springs, the length of the torque rod is equal to the wheelbase.

I have not, up to this moment, given the complete theory of the problem when the four wheels are braked. I shall do so now. Let us take a chassis in which all reactions, both front and rear, are taken by the springs. Let G be the center of gravity, H its height above the ground level, r the radius of the wheels, and P the weight of the car, as shown in Fig. 20.

Let F be the inertia force due to brake application. It is contrary to the sum of f_1 and f_2, which are the reactions to braking at the points of contact with the ground. These three forces set up an overturning moment $F\ H$ which, on brake application, introduces the variation of vertical reaction on the tires.

The overturning moment, taken with respect to the plane of the suspension system is $F\ (H - r)$. To make the overturning moment equal to zero, relative to the suspension plane, H must be equal to r; that is, the suspension should be in the same horizontal plane as that of the center of gravity. Usually, $H = 2r$. Therefore, on the average car, the overturning moment is $FH / 2 = C$, and, as F is a maximum of 0.60 of the weight of the car, and H = about 0.700 meters, we have $C = 0.21P$. This moment corresponds to a slight dipping of the front of the car on brake application; it is admissible dipping, but better made to disappear.

Let us now introduce a torque rod of finite length on the rear axle. We are now introducing a new couple P_2E_2, which alters the equilibrium.

As shown in Fig. 21, let l_1 and l_2 be the horizontal distances of the axles from the center of gravity. Let E_1 and E_2 be the distances of the extremities of the torque rods from the center of gravity. The correcting moment due to the rear axle is: $E_2P_2 = C_2$.

where
$$P_2 = f_2 \frac{r}{l_2 \pm E_2} \tag{6}$$

therefore
$$C_2 = f_2 \frac{\pm E_2\, r_2}{l_2 \pm E_2} \tag{7}$$

In the same way, the front correcting moment is:

$$C_1 = f_1 \frac{\pm E_1\, r}{l_1 \pm E_1} \tag{8}$$

We can conclude, therefore, that the correcting couple of a torque rod directed toward the center of gravity has a beneficial effect only if the rod is attached behind the center of gravity in the case of the rear axle, and in front of the center of gravity for the front axle.

In the transverse parallelogram type of front independent springing of the Peugeot car, this correcting couple is small, but it is directed in the right direction.

Care must be taken, in the case of the front, or steering wheels, not to attach the torque rod on the side opposite to the center of gravity with respect to the wheels and, in particular, with attachment in front of the front wheels. As an example, I am giving here the calculations made for the additional dipping moment in the first design of the Dubonnet car, based on the following data, as illustrated in Fig. 22.

The radius of the torque rod was assumed to be 0.250 meters.

One-half of the weight of the car is assumed to be supported by the front axle.

The adherence coefficient is 0.60.

The radius of the braked wheel assumed is 0.350 meters.

The height of the center of gravity assumed is 0.600 meters; and the wheelbase is 3.500 meters. Hence we have:

$$C_1 = \frac{P}{2} \times 0.60 \times \frac{0.350}{0.250} \times \left(\frac{3.500}{2} + 0.25 \right) = 0.84P \tag{9}$$

The braking disturbance passes therefore from 0.21 to 0.21 + 0.84 = 1.05. It is therefore multiplied by about 5.

The question is, in consequence, while designing a parallelogram-type front-axle, whether it is possible to eliminate completely the rearing or dipping moment. This is, of course, quite possible, on the condition of making the front torque-rod sufficiently long. It is also possible, on cars fitted with rigid front wheels, where all the torques are taken by the axle itself (Sizaire, Lancia, Broulhiet), to obtain equilibrium by placing a reaction rod on the rear axle which would be shorter than half the wheelbase.

This justifies the system adopted by Hispano for the transmission to the rear axle with reaction relays in case the four wheels are braked. (In the past I had rightly criticised the same system in the case of power transmission by the rear wheels and braking on rear wheels only.)

To sum up, from the point of view of the longitudinal stability, cars fitted with front axles of the classical type, excepting those having a reaction triangle, are not in a correct state of equilibrium as regards dipping moment, to the extent of a torque value of 0.20 P, where P is the weight of the car. This error causes a tolerable dipping of the front of the car during braking.

By fitting more flexible springs on composite axles, the bad effect due to the above error is accentuated. A faulty disposition of reaction levers would still more accentuate this defect, and this must be avoided. On account of this, those systems of independent front wheels having a single plain longitudinal lever with joint in front of the wheels, are bound to disappear.

[8] See *Revue de l'Industrie Minérale* (France), Sept. 1, 1921.

Let us now examine the conditions of overturning on road curves.

The overturning moment is proportional to the height of the center of gravity above the longitudinal axis of oscillation of the car. As in a correct independent springing, the oscillation axis is on the ground level, the overturning moment is twice as high. (At the moment, on passenger cars, the longitudinal oscillating axis, or rolling axis, is at about 0.350 meters from the ground, while the center of gravity is at 0.700 meters from the ground.)

To reestablish equilibrium, the spring torque recalling the car onto the level must be doubled. It is known that this couple is inversely proportional to the flexibility of the spring and proportional to the square of the transverse distance between the springs, on a car fitted with springs of the ordinary type.

I shall now demonstrate a special feature of independent-wheel systems. Let a_1 be the distance between the spring attachment and the wheel plane, and a_2 the distance between the spring attachment and the center line of the car. Let f be the flexibility of the spring; that is, the ratio between the static deflection and the weight. See Figs. 23 and 24.

For an ordinary axle, the righting couple is proportional to a^2_2/f.

For an independently sprung wheel axle, in an overturning movement, the displacement of the point A is increased by the fact that the joint A on the wheel has approached the ground.

For an ordinary axle, for an overturning angle α, the additional force in the spring is $\alpha a_2 = F$.

In independent wheels, it is equal to $\alpha(a_2+a_1)$.

In the ordinary axle, the righting couple is one and only, and applied to the car by the center of the spring. Its value is therefore $Fa_2 = \alpha a^2_2$.

In the composite type of axle, the righting couple of the wheel itself must be added. The wheel is always, by definition of correct independent springing, geometrically parallel to the center plane of the car. This additional righting moment has a_1 as a lever arm and its force is $(\alpha/f)\ (a_2+a_1)$.

The moment applied to the car is $(I/f)[\alpha(a_2+a_1)a_2]$.

The torque applied to the wheel plane is $(I/f)[\alpha(a_2+a_1)a_1]$.

Adding these two, we have:

$$(I/f)\ [\alpha(a_1+a_2)^2] = (I/f)\alpha I^2 \qquad (10)$$

The following theorem can now be enunciated. It is that, in independent springing, the value of the righting couple due to the springs must be calculated just as if the transverse springs had the same length as the track of the car.

The consequences of this theorem are of importance. The distance between the longitudinal front springs of an ordinary axle is about equal to half the track of the car. For an equal flexibility, the righting couple on a composite axle is four times that of an ordinary spring. As the overturning moment is doubled, nothing is altered as regards lateral stability of the car, if the flexibility of the springs is doubled.

For the rear axle, in the ordinary system, the distance be-

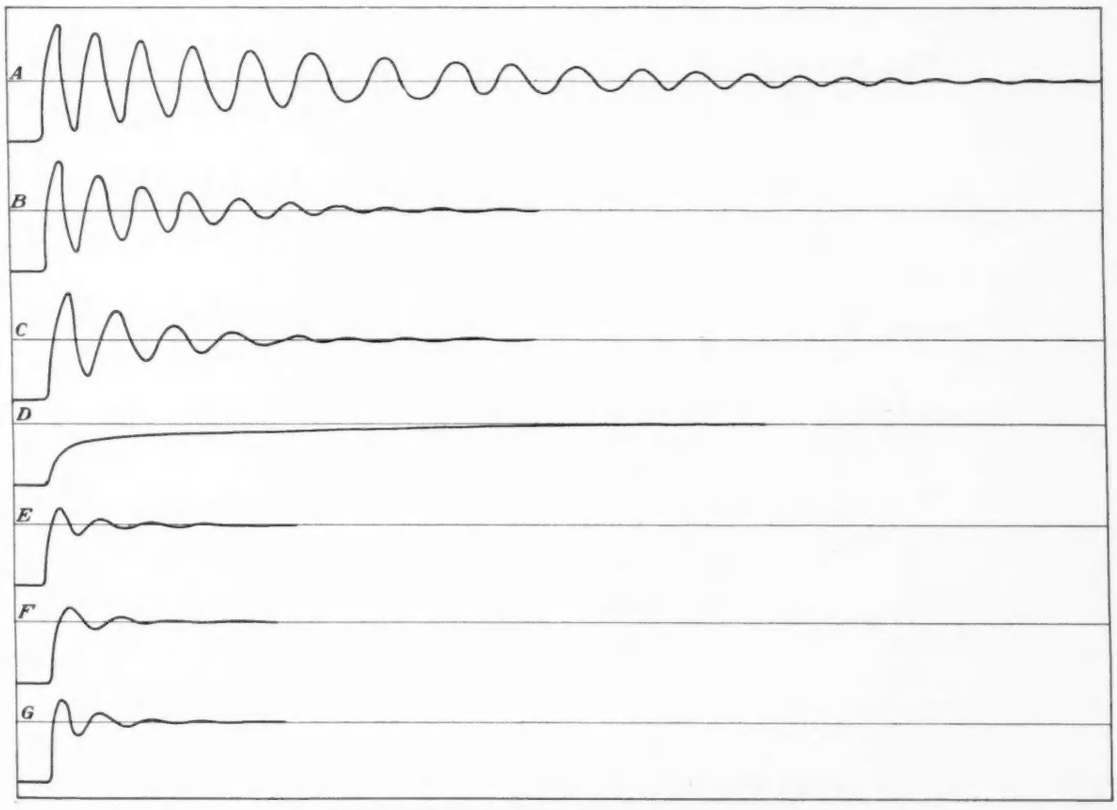


Fig. 19—Study of the Oscillations of a Rear Leaf-Spring of a Car

Curve *A*, Broulhiet frictionless, roller-equipped spring; Curve *B*, the same spring with rollers removed; Curve *C*, production spring of the same flexibility built up on an hypothetical basis—weight greater; Curve *D*, Broulhiet roller-equipped spring with "chatterton" between the leaves, tested 65 hr. after filling; Curve *E*, the same spring after 100 oscillations; Curve *F*, the same spring after 1000 oscillations; Curve *G*, the same spring after 6000 oscillations.

tween the springs is usually five-sevenths of the track; the square of this fraction is approximately equal to one-half.

On the rear, an independent-wheel system can therefore be substituted for an ordinary type of axle without modifying the lateral equilibrium, on the condition that the flexibility remains strictly the same. If the rear flexibility is increased, the overturning tendency is increased. The lateral

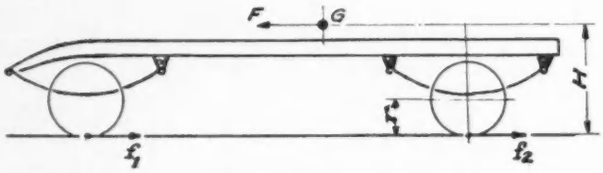


Fig. 20—Diagram of a Chassis in Which All Reactions, both Front and Rear, Are Taken by the Springs

The center of gravity is indicated at *G*, its height above ground level at *H*, the radius of the wheels at *r*. In the text, the weight of the car is represented by *P*. The inertia force due to brake application is indicated at *F*. It is contrary to the sum of f_1 and f_2, which are the reactions to braking at the points of contact with the ground.

stability therefore remains unchanged when, with front independent springing, the flexibility of the springs is doubled.

I wish to say a few words on a problem which is always much discussed in a new design of a car, and never after test. I mean the transverse inclination of the front-axle pivots, and the determination of the optimum impact in respect to the steering and the braking.

Before the era of front-wheel brakes, steering knuckles were often designed as in Fig. 25, with a very large value for *a*, often greater than 100 mm. With the advent of front-wheel brakes, inclined pivots were used, with a distance *a* varying from 0 to 60 mm. See Figs. 26 and 27.

This inclination gives rise to various remarks:

(1) It introduces friction in the pivoting motion, on account of the torque, even if the impact is zero. This torque is reduced to zero only if the pivot is vertical and on the plumb-line of impact, as in Fig. 28, with the pivot inside of the wheel.

(2) If the impact is zero, due to reducing the dimension *a* to zero, this never eliminates the effect on the steering gear of lack of braking adjustment.

(3) As the braking adjustment becomes equal through wear, an increase of dimension *a* has no effect on the steering qualities of the car during braking.

I have tried, with a telescopic slipper-type independent-springing the following dimensions for *a*; 28, 52, 73, 90 and 100 mm. I found no difference in the steering stability under braking effect.

It is convenient, in a slipper-type independent-springing to have a clearance between the pivot and the wheel. This gives an impact value of about 100 mm. This value, which may seem high, is sanctioned by the many years of experiments the Lancia type of car has been effecting, and the excellence of whose braking is well known. The same impact figure is used in the Dubonnet suspension system.

I should like to point out that the transverse-parallelogram system, in which the ordinary type of steering knuckles can be used, has generally an impact figure of from 25 to 60 mm.

The variation of this dimension *a* and its increase are used principally in cases in which shimmy is found to take place in a classical axle and attempts are being made to correct it. In fact, the inertia of the wheel to pivoting can be adjusted in this way. The diminution of the dimension *a* has not therefore as an object the equalization of the front-wheel braking-effect.

Having summed up both the longitudinal and transverse equilibrium of the car, I shall now pass the existing mechanisms in review.

We are now leaving direct theory behind, to examine the facts. We are therefore leaving the domain of Science to enter that of Invention.

About 60 types of mechanisms have been invented, which I shall divide into three groups.

(1) *Slipper-Type.*—Lancia and Broulhiet.

(2) *Transverse Parallelogram.*—Sizaire, Peugeot, Mase and Broulhiet.

(3) *Longitudinal Parallelogram.*—Porsche.

The Dubonnet solution can be kept on account of the interesting suspended steering mechanism which it embodies, as the error in longitudinal equilibrium can be corrected. The above solutions are well known. As a complement, I have given, in an Appendix, a description regarding the solutions that I myself have proposed.

I shall now, in a very general way, discuss some of the criticisms and objections I have received. To begin with, from tire manufacturers. For them, apparently, the solution of the problem lies in the tire itself, that is, large sizes and low pressures. But large sizes and low pressures do not always adapt themselves to all types of steering mechanisms. Then they say: "Let us adopt independent springing so as to be able to use larger tires and correct the steering." But, among the large number of types of independent springing, the majority have a greater steering reaction than does the classical type of front axle.

The tire manufacturer then changes his argument, which is entirely commercial, and says: "All good independently sprung cars having balloon tires owe their qualities to the tires and not to the independent suspensions."

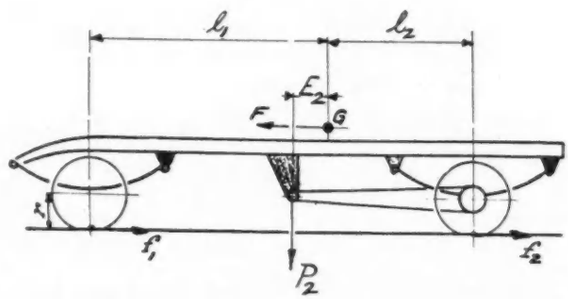


Fig. 21—Diagram of the Chassis Shown in Fig. 20 after Introducing a Torque Rod of Finite Length on the Rear Axle

This introduces a new couple, P_2 E_2, which alters the equilibrium.

Excessive tire wear now takes place. It is attempted now to establish an interconnection between the geometrical imperfections of the composite axle and the tire wear, and both the variation of the track and the variation of the toe-in of the front wheels are also involved. This question is important and bears investigation. Since a short time ago, there are continual complaints, in Europe, regarding excessive tire wear. The fact that the most curious and most marked cases of wear are to be found on the rear axle of light trucks, exempts the composite front axle as solely responsible for the excessive wear.

The slight variation in the toe-in of composite axles is now involved. It is sufficient to remember that, on the classical type of front axle, the wheels are mounted with a toe-in varying from 8 to 16 mm. measured on the tire bead, and until the toe-in does not reach the figure of ± 8 mm., it cannot be said that the independent-wheel system has brought forward a harmful novelty with respect to the classical type of axle.

Experience has shown that it is always difficult to adjust a parallelogram type of axle, however theoretically correct the design be for no variation in toe-in. The defect has only very small importance if it remains small; within ± 6 mm., measured on the bead, for vertical displacements between minus 40 mm. and plus 80 mm., on the condition that the toe-in variation be the same for each wheel; otherwise, a variation in the car steering would result at each displacement of the front suspension. For this reason it would appear necessary to have a suspended steering rod in a parallelogram system. In this way, the steering mechanism should comprise a track rod, split in the middle, with the connection to the steering box at this point, and not on the knuckle of one of the wheels. Nevertheless, with good workmanship, a good result can be attained, even with this latter system.

Much importance is attached to the variation of track in transverse parallelograms, in so far as it refers to tire wear. I should like to point out that, with balloon tires, a variation of 16 mm. does not cause the tire to slip on the ground when the wheel is stationary. I have covered many miles on a two-legged type split front-axle in which there was a large variation in track with a set of Michelin 12 x 45 tires (Peugeot 201 test car) and have never noticed any excessive wear on the tires.

Sizaire has based the study of his independent springing on a parallelogram having unequal arms, so as to obtain a constant track, neglecting the lack of consistent parallelism, and he admitted an error of 3 deg. in the parallelism to obtain a rigorously constant track. He justified his action by his experience with this system.

Julien has made the frontal displacement-diagram for a parallelogram system having unequal arms, and has brought the error in parallelism down to 1 deg., with a track variation of 2 mm. only. I have utilized this diagram in my latest parallelogram system. I give here a description of Julien's diagram, shown in Fig. 29.

Taking the lower arm AB and three points a, b and c, on the vertical trajectory of tire impact, the three corresponding points a_1, b_1 and c_1 on the top joint of the knuckle can be deduced. Place the center of rotation of the top lever at the center point of a circle passing through the three points a_1, b_1 and c_1.

Excessive wear in tires on composite axles is often due to the driver during the tests; the facility of the car to steer in a short space is often abused. As soon as the car is driven in a normal manner by a customer who has paid for his car and who pays for his tires, wear once again becomes normal.

There now remains to investigate the parasitic effects inherent to all types of solutions.

The best result and complete immobility of the steering mechanism can be obtained only by eliminating any constant friction in the front suspension; it is therefore necessary to fit shock absorbers having a damping effect proportional to the speed, and frictionless springs.

In the slipper-type solution, it would be harmful to have any appreciable friction. In the one I designed, I was forced to use frictionless ball bearings to reduce the friction to zero. The system would lose all its qualities if the balls were to be removed. Another advantage, of the practical order, in using balls, is the small amount of lubrication upkeep necessary.

In the Porsche car where the efforts are very high on account of the shortness of the parts, to eliminate the friction in the bearings, needle rollers had to be resorted to.

The perfect result obtained in the slipper type and with the Porsche system of levers is entirely dependent on the reduction of friction in the mechanical movements. Suppression of friction is only a question of practical mechanical fitting, and it is only necessary to have the will to do it. On the contrary, in transverse parallelograms, this problem does not need to be dealt with, not even in the initial stages. The bearings are far apart, the efforts small, and the pins usually mounted on "Silentbloc" of small diameter. For this reason fitting is quick, and of no special difficulty.

The parallelogram takes advantage of the large sizes of all the parts, which allows high-resistance design with a low weight. The last parallelogram I designed on a car weighing

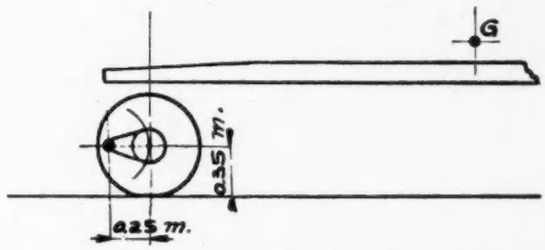


Fig. 22—Diagram to Illustrate the Calculations Made for the Additional Dipping Moment in the First Design of Dubonnet Car

1400 kg. weighed 10 kg. less than the actual production axle.

The same reason for the choice of light materials renders the transverse parallelogram more bulky, and requires a large number of joints.

The slipper type of independent springing has uncontested advantage on all parallelogram types built up to the present. This advantage resides in the fact of a perfectly rectilinear trajectory on the road.

The Reception Accorded by Research Organizations.—In France, research organizations are now passing through those same states of mind through which I myself passed as my studies went on. I understood the need for independent wheels when I realized the impossibility of removing certain resonances from vehicles by the redesign of the suspension only. It was as a result of numerous trials with suspension by counter-accelerators (springs of variable flexibility) and sensitive springs that I confirmed that the vibration of the front axle and the wheel rebound depended on conditions altogether different from those that could be caused by a strictly elastic device. I arrived at two conclusions:

(1) The need for separating the operation of the rotating system from that of the suspension proper, and

(2) The need for studying the unexplained resonances of the front axle with the purpose of improving the suspension.

I have published a study of this subject in the *Revue l'Industrie Minérale,* of Saint-Etienne, and I have read with interest, in the lectures of M. le Commandant Prevost, given by him at the *Centre d'Instruction Militaire de Fontainebleau,* that I was the first to have separated these two functions of suspension. Commandant Prevost's remark, of which I was not aware at

that time, attracted my attention and showed me how much more important the question was than I myself had thought. I am therefore indebted to Commandant Prevost for having clearly shown me that the path which I was following, because of mere deduction from test work, was the only true path, generally speaking.

As I never experiment except on a subject on which theoretical calculations have been made, I again immersed myself

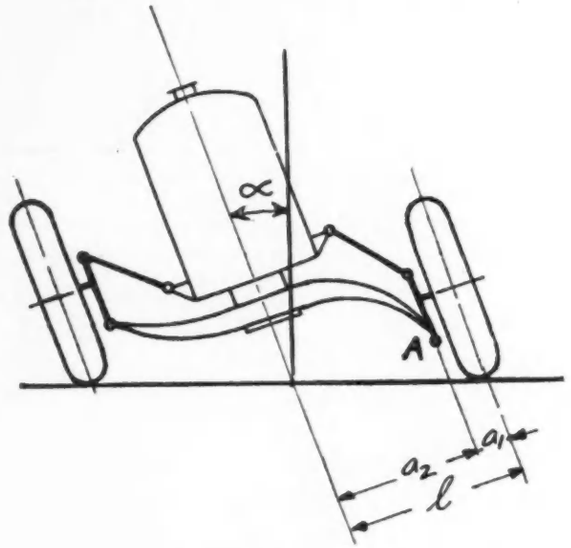


Fig. 23—Overturning Moment Illustrated

in mathematical analysis; on the one hand, I presented my study on gyrostatic resonances before the Société des Ingénieurs Civils de France and I conceived the theory of double suspension, which I presented to the Société des Ingénieurs de l'Automobile de France, through numerous communications.

The scientific investigation of these two questions spread before me in all its extent the problem of resonance, and I constituted myself the advocate of the idea that the study of the subject, vehicle locomotion, depends entirely on the theory of resonance. This was the thesis that I upheld in connection with a competitive examination before a board of which Robert Peugeot was a member. A few days later, Robert Peugeot called upon me to cooperate in improving his production and introduced me to his engineers at his Sochaux factory. As a result of Peugeot's success, André Citroèn called upon me for the same purpose.

The reception accorded to this theory of resonance has been of the same type in the research organizations of these two large companies. It indicates a distinct state of mind, peculiar to French automobile engineers holding executive authority. Their lack of understanding causes considerable hindrance to progress.

The general theory of resonance, by which is understood gyrostatic resonance, offers to research organizations a solid foundation for arriving at two results: (*a*) the complete disappearance of steering disturbances; and (*b*), the complete disappearance of the vibrations of the vehicle due to the road. A third result, roadability, appears automatically. That has been obtained in a different way in certain designs but with a front suspension not acceptable today.

Good results cannot be obtained except by conforming to the rules laid down by the theory, which is therefore exclusive. One must submit to it or resign. In each of the two companies, Peugeot and Citroèn, I have found quickly engineers sympathetic to the new theories and with an understanding of their exclusiveness.

The question may be summarized as follows. I required:

(1) Chassis construction rigid against torsion, but without decrease in the modulus of vertical inertia.

(2) The construction of a split axle embodying the conditions of parallel displacement.

(3) The construction of sensitive springs and the use of shock absorbers from which all friction constants were excluded.

(4) As a means for eliminating fender resonance, which sometimes persists even on rigid chassis, I required independence of the fenders.

If research organizations had believed in the need for the first three principles, it would have been possible for them to conceive models filling these conditions.

In the first company, the need for rigidity was at first not believed in and objections were made to accomplishing it; such as "heavier, more expensive, less robust." The models, except those constructed under my direct control, were not provided with tubular chassis or with frictionless shock absorbers. Result: the first defeat.

A body designed as a parallelogram, with friction at the joints, on a conventional chassis, will exhibit torsion at the front of the body, movements of the steering wheel and the impossibility of incorporating in the front axle friction higher than 15 kg. per wheel without causing inadmissible vibrations. Under these conditions, the damping of the wheel rebound was inadequate on bad roads; steering accuracy was, however, excellent.

Fortified by this experience, Hogard and I then studied the layout of the Peugeot 201-C type, which embodied a reinforced chassis, a front spring with rollers between the leaves and shock absorbers with damping proportional to the speed of the oscillations. The result was thought adequate by the chief executives, Robert Peugeot and Mattern, and the rigid chassis was held in reserve.

At the time when the cars of the first series, exhibited at the Salon de l'Automobile at Paris in 1931, were built, the following thoughts were expressed by some engineers, opinions which have not changed my feeling of sympathy for

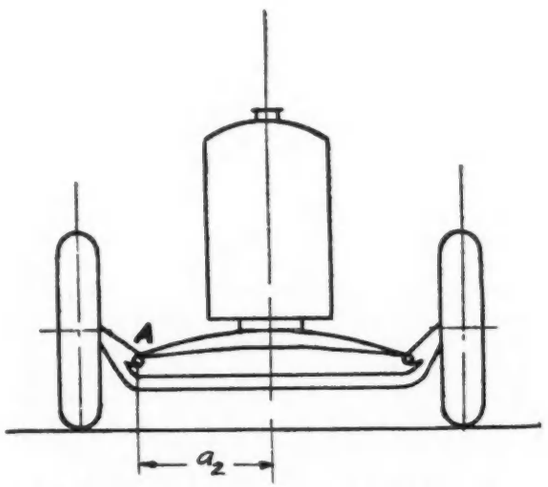


Fig. 24—Diagram Supplementing That in Fig. 23

them because of the intense sincerity with which the opinions were delivered:

"Independent-wheel progress is useless to the public; the market does not require it. The theory of rigidity has more disadvantages than advantages, and I do not believe that the rigid chassis will ever see the light of day commercially."

The rigid chassis was then introduced, at the time of the design of the 301-C car, in June, 1933. It has just been accepted for the construction of all the touring cars of the Peugeot company and it has now replaced the former conventional reinforced chassis of the 201-C model. These facts have brought me the experience of having lived through the business, and enable me today to hold my own against objectors whom I meet in other companies.

In another company, the lack of understanding on the part of the manager took a more curious turn. Here is what his engineering director said to me one day:

"You are not yet ready; the Boss has asked me to help you; it is not my opinion; I do not wish to know the means you use. When you shall have done all you can and will say to us: 'This is the best I can do,' we will examine whether there are any positive results, and, if so, we will be able to take up your task from where you have left off and get the best of it, if there is something in it."

This policy was followed with great firmness and is still being followed. It is a wrong policy from the viewpoint of progress and leads nowhere. You can't tell a research man that he has finished until he is dead.

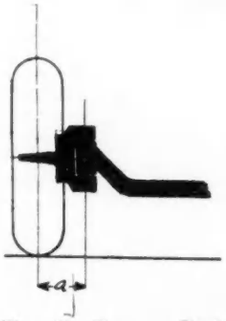


Fig. 25—Former Design of Steering Knuckle Having a Very Large Value for Dimension *a*

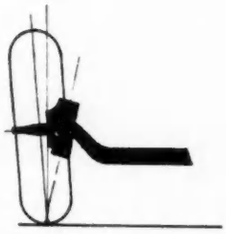


Fig. 26—Another Type of Steering Knuckle with Dimension *a* Having Zero Value

Meanwhile, the research organization, whose duty it was to suggest a model, has done nothing personally, and has limited itself to witnessing experiments the conditions of which, according to its pronouncement, it does not wish to know. The research organization shows itself incompetent to produce anything on a proposition on which the head of the industry has decided to embark.

In writing this history on behalf of progress, I emphasize that the science of locomotion has not made a single backward step; more than that, it has not yet reached the limit of its possibilities and when the designs based on the principles enunciated will have been fully realized, it will become necessary to design the rear suspension for deflections under load of the order of 500 mm., with a return to the installation of counter-accelerators or of stabilizers with regulation more or less according to load.

In another company, a course of contrary ideas has been manifested which explains the absence of personal studies on independent wheels. It is claimed that the same results are obtained with an improved ordinary axle.

This distinction must be made: a poor split axle is worse than a good conventional axle. That is all that poorly directed experimentation can be made to confirm. Moreover, there is a possibility along that line, on condition that roads continue to improve as has been the case in France for ten years. Large deflections under load, that independent wheels have shown possible at the front of vehicles, combined with good shock absorbers with damping proportional to the speed, truly

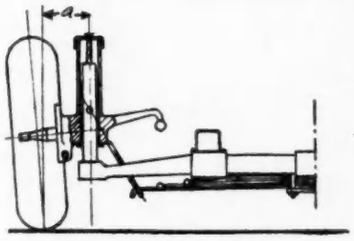


Fig. 27—Steering Knuckle Having Still Another Value for Dimension *a*

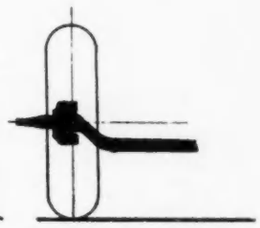


Fig. 28—Steering Knuckle in which the Torque is Reduced to Zero only if the Pivot is Vertical and on the Plumb Line of Impact, with the Pivot Inside of the Wheel

permit the adherence of the wheel to the ground. One is justified, then, in asking one's self if it is useful to guard one's self against gyroscopic effects, considerably decreased, since the simple conventional axles remain parallel to the ground. The objection that super-flexible springs cannot oppose braking reactions can be met by the use of reaction rods. This is a path toward improvement on which Alfa-Roméo has, rightly, entered. But the solution is applicable, theoretically, only to good roads.

The consequence of researches toward the mechanical embodiment of independent wheels will be, moreover, solutions as light and as inexpensive as the conventional axle. There are, moreover, designs that have reached that goal.

To summarize, the way in which research organizations have received independent wheels shows how desirable it is from the viewpoint of the most fruitful method to bring about the assemblage of principles by means of all suitable experimentation and so to prepare a solid foundation for inventors.

This explanation has seemed to me useful in the history of independent-wheel development, for it is only the repetition of what occurred with front-wheel brakes, thoroughly developed at the Isotta-Fraschini company, among others, in 1906, and which did not achieve a period of widespread distribution until a number of years later.

The initiative of the Peugeot company, due to the decision of its chiefs, who had judged correctly, is beneficial to progress. The widespread commercial experience achieved by Peugeot shows clearly that there exist no obstacles to the generalization of independent front wheels. The advantages that users draw from them are such that it can be foreseen that in several years the general use of independent front wheels will be an accomplished fact; as few vehicles without independent front wheels will be seen as vehicles without front-wheel brakes.

Reception Accorded by the Public.—At whatever time split axles have been presented to the public, this type of design has never decreased the value of the car. Amédée Bollee suffered no grief from his split front exle with two parallel transverse springs, ancestor of all present designs where the springs play the role of rods.

If I run through history, I find successively: Sizaire, with a slipper type, in 1906; Gauthier, with an oscillating lever, in 1920; and Beck, with a rod, a mechanical embodiment of the Gauthier design. Then came Ram and Lancia; then Sizaire, with the constant track; and then the crowd.

The public has always been favorable to independent wheels. Two cars, moreover, have experienced a sure success: Lancia and Sizaire, especially the first. There exist old vehicles of these two builders that have run, under the same conditions as those of the best classical makes, 200,000 or 300,000 km. in ten years to the satisfaction of their owners. These two systems have, unfortunately, not been incorporated on vehicles of widespread distribution. The public would have taken all that would have been offered to it.

The first car with independent front wheels to be placed on a quantity-production basis is the Peugeot 201-C model, followed by the 301-C model of the same company. The guarantee of well-planned production based on correct principles has procured for that company a brilliant success, which is no longer disputed, except in the research organizations of its competitors.

If one sets aside the comments of ill-tempered competitors the success of Peugeot is complete and is shown in three steps. The 201-C model, with reinforced chassis; the 301-C model, with rigid tubular chassis; and the 201-C model with rigid tubular chassis, as well as the generalization of independent wheels and the rigid chassis in the building of all touring cars.

This company has found in this design a new element of prosperity. I do not say the sole element of prosperity, for this company, in its splendid Sochaux factory, possesses all the trump cards which affect production facilities under conditions of remarkable economy, and would have been able with the conventional model to continue to occupy an outstanding position on the market.

Mr. Robert Peugeot, one of the principal spirits of this company, told me very recently that he did not yet know of any complaint on the part of the public on the subject of independent wheels, and that there was no necessity at present for modifying anything in a system which had shown itself to be perfect so far as contact with the public was concerned.

I clearly could not hope for more complete satisfaction, but I must think that all is relative in large-scale production and that Peugeot, knowing no rivalry, is not at the limit of progress in its present state.

The limit of possibilities cannot, moreover, be reached except through commercial competition on the basis of quality. This has always been the law of automotive progress in the past.

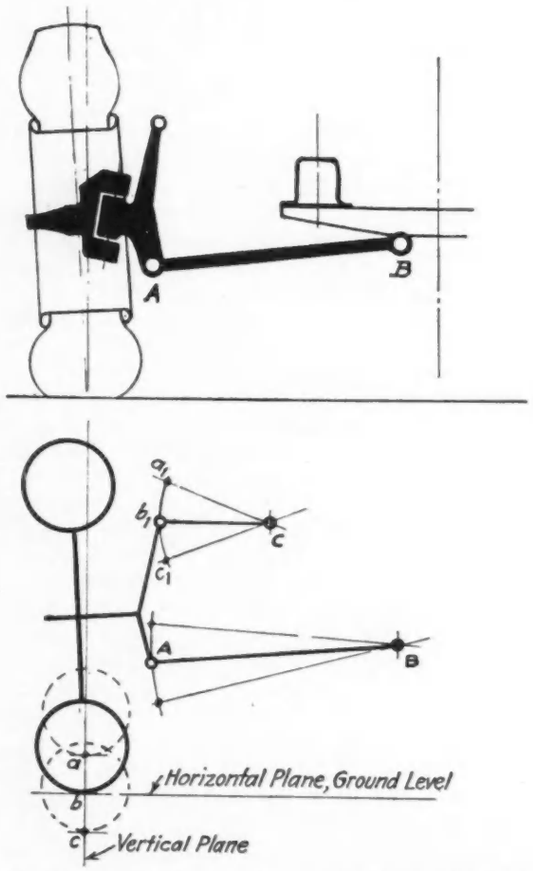


Fig. 29 — Parallelogram Type of Independent Front Wheels—Julien's Displacement Diagram

Appendix 1

Shock Absorbers With Damping Proportional to Speed of Oscillations—Broulhiet System

Shock Absorbers in Vehicle Suspension: Introduction.—In the classical differential equation of a pendular movement:

$$M\frac{d^2x}{dt^2}+f\frac{dx}{dt}+Cx=0 \qquad (1)$$

M is the mass and C is the bending constant or rating of a spring.

The terms $M\frac{d^2x}{dt^2}$ and Cx are reversible forces, an inertia force and an elastic force.

The term $f\frac{dx}{dt}$ is an energy-destroying force, a damping force introduced into the classical equation in the shape of a force proportional to the speed with the object of facilitating the integration of the equation and the study of the movement.

It has seemed to us interesting to attempt to design a shock absorber meeting this analytical conception and to apply such a device to the damping of oscillations in the springs of a car. In order not to deviate from theoretical concepts we have been led to use frictionless springs in our tests; that is, coil springs or leaf springs with rollers interposed between the leaves.

Mechanical Realization of Damping Proportional to Speed.—Let us consider an hydraulic shock absorber comprising a piston displaced in a cylinder and causing the liquid to flow through orifices. See Fig. 30.

Let Ω be the cross-section of the piston; S, the cross-section of the orifices; and h, the pressure of the liquid. The speed of the passage of the liquid through the orifices S is: $V=\sqrt{2gh}$. The damping force is: $F=h\Omega$, a force that we suppose proportional to the speed of piston displacement; hence,

$$\Omega h=f\frac{dx}{dt} \qquad (2)$$

The volume of liquid displaced by the piston is:

$$Q=\Omega\frac{dx}{dt} \qquad (3)$$

The flow of liquid through the orifices S is:

$$Q = SV = S\sqrt{2gh} \tag{4}$$

hence

$$S\sqrt{2gh} = \Omega\frac{dx}{dt} \tag{5}$$

Dividing Equations (2) and (5) member by member:

$$\frac{S\sqrt{2gh}}{\Omega h} = \frac{\Omega}{f} \tag{6}$$

$$S = \frac{\Omega^2}{f\sqrt{2g}}\sqrt{h} = K\sqrt{h} \tag{7}$$

The cross-section of the orifices should therefore be proportional to the square root of the pressure.

We have realized such a device in a piston valve, represented in Fig. 30. The displacement of this piston valve is controlled by a spring having no initial tension in the position of repose; also, the stroke of the piston valve is proportional to the pressure exerted by the liquid. The openings are all of the same diameter. Their distance from the shoulder of the piston valve is 1, for the first orifice; 4, for the second; 9, for the third; 16, for the fourth, etc. Further, the cross-section of the passage is proportional to the equare root of the stroke of the piston valve. The damping force is independent of the temperature, thanks to the use of a very fluid oil and of thin-walled orifices of sufficient cross-section in the piston valve.

An hydraulic shock absorber so designed gives a damping proportional to speed. To determine the dimensions of the device we must know:

(1) The maximum damping force, on which depends the cross-section of the piston; and

(2) The maximum speed of the piston, on which depends the total of the cross-sections of the piston-valve openings.

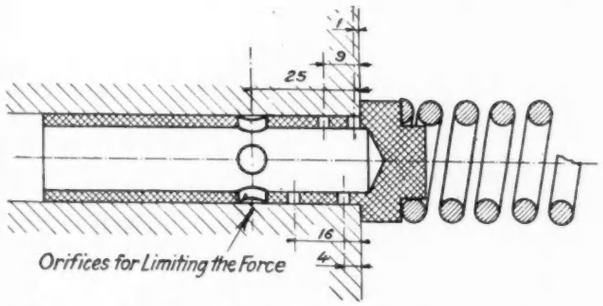


Fig. 30—Hydraulic Shock Absorber Comprising a Piston Displaced in a Cylinder and Causing the Liquid to Flow Through Orifices

Expansion of a Spring with Damping Proportional to the Speed.—Let us consider a frictionless spring, carrying a load of mass M, and let us calculate the position of the center of gravity of this mass with relation to a horizontal plane of a reference corresponding to a complete expansion of the spring. The load is the mass M, and the weight $P = Mg$.

The bending constant of the spring is C; and we have:

$$C = \frac{\text{Strength of the Spring in Kilograms}}{\text{Deflection in Meters}} \tag{8}$$

The damping coefficient is f; and we have:

$$f = \frac{\text{Damping Force in Kilograms}}{\text{Speed in Meters per Second}} \tag{9}$$

Let x be any position of the center of gravity, the compressed spring expands, x goes down and we have:

$$Cx - P = -f\frac{dx}{dt} - M\frac{dx}{dt^2} \tag{10}$$

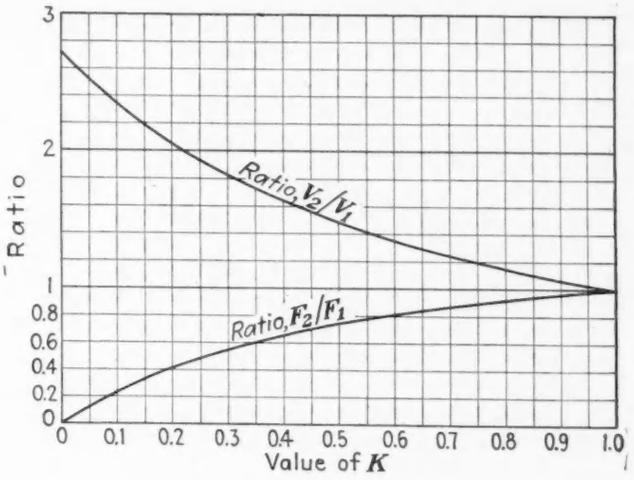


Fig. 31—Return Movement

V_1 and V_2 at maximum speed; critical aperiodic movement. F_1 and F_2 as maximum forces; damping movement, $K = (f/2\sqrt{MC})$.

The differential equation is:

$$M\frac{d^2x}{dt^2} + f\frac{dx}{dt} + Cx = P \tag{11}$$

Let us consider $x = e^{\omega t}$ (12) and $M\omega^2 + f\omega + C = 0$ (13)

We will study two cases: (a), $f^2 - 4Mc = 0$ (14)—that of equal roots and aperiodic critical damping; and (b), $f^2 - 4Mc = <0$ (15)—that of imaginary roots and with the movement damped.

Aperiodic Critical Damping

In considering case a, that of aperiodic damping, we have:

$$x = ae^{\omega t} + bte^{\omega t} + x_1 \tag{16}$$

$$\frac{dx}{dt} = \omega ae^{\omega t} + bt\omega e^{\omega t} + be^{\omega t} \tag{17}$$

$$\frac{d^2x}{dt^2} = \omega^2 ae^{\omega t} + bt\omega^2 e^{\omega t} + 2b\omega e^{\omega t} \tag{18}$$

Substituting these values in the differential equation, we find that $Cx_1 = p$ and that $\omega^2 = C/M$. The factor x_1 is the equilibrium position of the spring under the load P, and ω is the characteristic speed of undamped movement. We have also:

$$x = e^{\omega t}(a + bt) + x_1 \tag{19}$$

$$\frac{dx}{dt} = e^{\omega t}(a\omega + b\omega t + b) \tag{20}$$

$$\frac{d^2x}{dt^2} = \omega e^{\omega t}(a\omega + b\omega t + 2b) \tag{21}$$

Let us make $t = 0$. Then we have:

$$x = a + x_1 \tag{22}$$

$$\frac{dx}{dt} = a\omega + b \tag{23}$$

where a is the deviation of the equilibrium position of the spring, that is, the initial compression given to the spring.

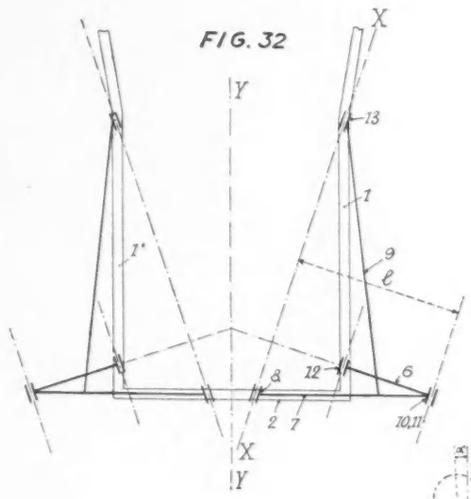
In Equation (23), $b = -a\omega$ for $\frac{dx}{dt} = 0$.

Equations (19), (20) and (21) become:

$$x = ae^{\omega t}(I - \omega t) + x_1 \tag{24}$$

$$\frac{dx}{dt} = -ae^{\omega t} - \omega^2 t \tag{25}$$

$$\frac{d^2x}{dt^2} = -ae^{\omega t}(\omega t + 1) \tag{26}$$



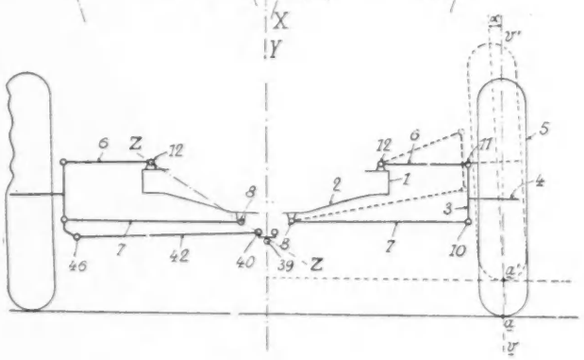


FIG. 33

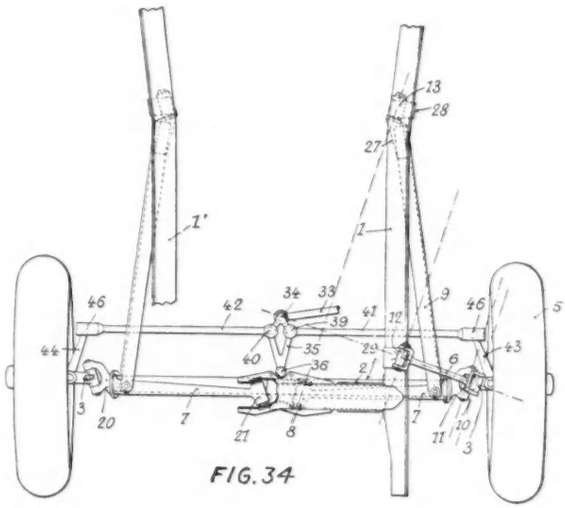


FIG. 34

FIG. 36

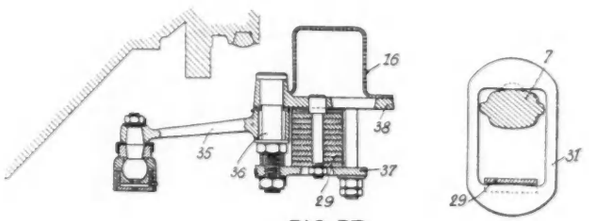


FIG. 37

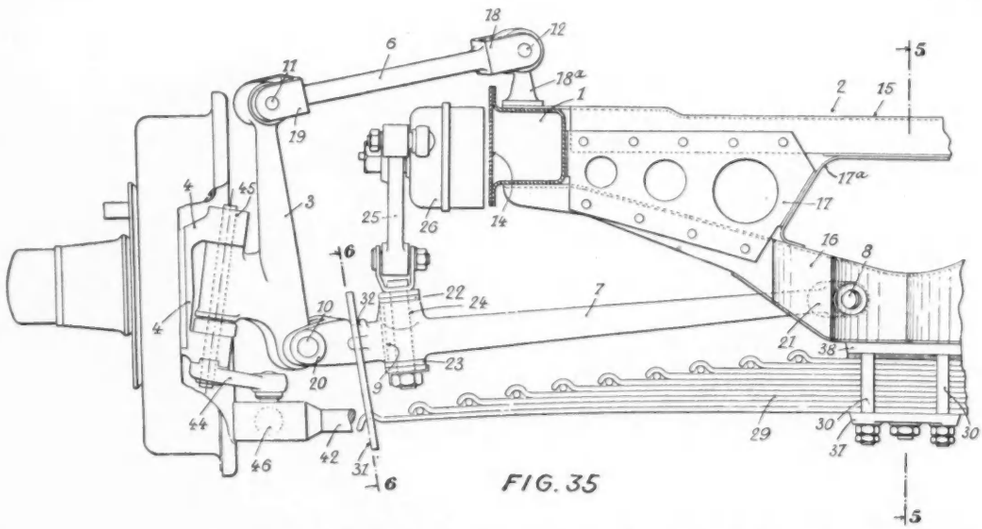


FIG. 35

Figs. 32 to 37—Broulhiet Parallelogram Type Independent Front-Wheel Diagrams

The maximum speed of return, V_1, is obtained for $\frac{d^2x}{dt^2} = 0$; or, for $\omega t = -1$. Hence,

$$V_1 = \frac{a\omega}{e} = \frac{a}{e}\sqrt{\frac{C}{M}}$$

$$e = 2{,}7183$$

$$\omega = \sqrt{\frac{C}{M}}$$

and a is expressed in meters.

The maximum damping force that the shock absorber could sustain is:

$F_1 = f\,V_1 = 2\sqrt{MC} \times V_1$, in kilograms.

From the point of view of construction, F_1 determines the cross-section of the piston of the shock absorber and V_1 determines the dimensions of the piston valve.

Damped Movement

In considering damped movement, case (b), let us take:

$$f^2 < 4MC \tag{27}$$

Let us make

$$K = \frac{f}{2\sqrt{MC}} \tag{28}$$

where K is a positive coefficient less than 1, and let ω' and ω'' be the imaginary roots of the equation:

$$M\omega^2 + f\omega + C = 0 \tag{29}$$

Hence we have:

$$\omega' = p + qi \tag{30}$$

$$\omega'' = p - qi \tag{31}$$

$$\omega' + \omega'' = 2p = -\frac{f}{M} \tag{32}$$

$$\omega'\omega'' = p^2 + q^2 = \frac{C}{M} \tag{33}$$

$$x = e^{pt}(a \cos qt + b \sin qt) + x_1 \tag{34}$$

$$\frac{dx}{dt} = -e^{pt}(aq \sin qt - bq \cos qt) + pe^{pt}(a \cos qt + b \sin qt) \tag{35}$$

Let us make $t = 0$. Then we have:

$$x = a + x_1 \tag{36}$$

$$\frac{dx}{dt} = bq + pa \tag{37}$$

where a is the deviation of the position of equilibrium of the spring; that is, the initial compression given to the spring. Hence,

$$b = -\frac{p}{q}a \text{ for } \frac{dx}{dt} = 0 \tag{38}$$

The equations then become:

$$x = ae^{pt}(\cos qt - \frac{p}{q} \sin qt) + x_1 \tag{39}$$

$$\frac{dx}{dt} = -a\frac{p^2+q^2}{q}\, e^{pt} \sin qt \tag{40}$$

$$\frac{d^2x}{dt^2} = -a(p^2+q^2)e^{pt}(\cos qt + \frac{p}{q} \sin qt) \tag{41}$$

The maximum and the minimum speed of return, V_2, are obtained for $\frac{d^2x}{dt^2} = 0$; or for

$$tg \times qt = -\frac{q}{p} \text{ and } \sin qt = \frac{-q}{\sqrt{(p^2+q^2)}} \tag{42}$$

Substituting this value in the expression for speed, we have:

$$V_2 = a\sqrt{(p^2+q^2)} \times e^{pt} \tag{43}$$

where t is determined from the relations:

$$\sin qt = \frac{-q}{\sqrt{(p^2+q^2)}} \tag{44}$$

$$t = \frac{1}{q} \text{ arc sin } \frac{-q}{\sqrt{(p^2+q^2)}} + n\pi \tag{45}$$

where $n = 0$ or a whole number.

Let us make

$$\omega^2 = \frac{C}{M} \tag{46}$$

$$p^2 + q^2 = \frac{C}{M} = \omega^2 \tag{47}$$

$$p = -\frac{f}{2M} = -\frac{f}{2\sqrt{MC}} \times \sqrt{\frac{C}{M}} = -K\omega \tag{48}$$

$$q = \pm\sqrt{(\omega^2 - p^2)} = \pm\omega\sqrt{(1-K^2)} \tag{49}$$

Bringing into the equation:

$$\sin qt = \frac{-q}{\sqrt{(p^2+q^2)}} \tag{50}$$

we have:

$$\sin \omega\sqrt{(1-K^2)} \times t = \frac{\pm\omega\sqrt{(1-K^2)}}{\omega} = \pm\sqrt{(1-K^2)} \tag{51}$$

and

$$\omega t = \frac{1}{\sqrt{(1-K^2)}} \times \text{ arc sin} \pm \sqrt{(1-K^2)} + n\pi \tag{52}$$

Bringing into the equation:

$$V_2 = a\sqrt{(p^2 + q^2e^{pt})} \tag{53}$$

we have:

$$V_2 = a\omega e^{-K\omega t} \tag{54}$$

The first maximum of speed is obtained from the expression:

$$\omega t = \frac{1}{\sqrt{(1-K^2)}} \times \text{ arc sin } \sqrt{(1-K^2)} \tag{55}$$

Developing arc sin in a series, we have:

$$\omega t = 1 + \frac{1}{6}(1-K^2) + \frac{3}{40}(1-K^2)^2 + \frac{15}{336}(1-K^2)^3 \ \ldots\ . \tag{56}$$

Let us compare this value V_2 with the value V_1, found for aperiodic critical damping. Then we have:

$$V_1 = a\omega e^{-1} \tag{57}$$

and

$$\frac{V_2}{V_1} = \frac{a\omega e^{-K\omega t}}{a\omega e^{-1}} = e^{1-K\omega t} \tag{58}$$

Comparing the damping forces, we have:

$$F_1 = f\,V_1 = 2\sqrt{MC} \times V_1 \tag{59}$$

$$F_2 = f\,V_2 = K \times 2\sqrt{MC} \times V_2 \tag{60}$$

$$\frac{F_2}{F_1} = K\frac{V_2}{V_1} = K \times e^{1-K\omega t} \tag{61}$$

In Equation (56) the value of K is included between 0 and 1. Table 1 and the curves in Fig. 31 are deduced.

Table I—Appendix I

K	ωt	$K\omega t$	$1 - K\omega t$	$\frac{V_2}{V_1}$ $e^{1-K\omega t}$	$\frac{F_2}{F_1}$ $Ke^{1-K\omega t}$
0	$\frac{\pi}{2}$	0	1	2.7183	0
0.1	1.48	0.148	0.852	2.345	0.2345
0.2	1.40	0.280	0.720	2.054	0.4108
0.3	1.325	0.397	0.603	1.829	0.5490
0.4	1.265	0.506	0.494	1.639	0.6550
0.5	1.205	0.602	0.398	1.490	0.7450
0.6	1.160	0.696	0.304	1.358	0.8140
0.7	1.112	0.778	0.222	1.249	0.8740
0.8	1.070	0.856	0.144	1.155	0.9250
0.9	1.033	0.930	0.070	1.073	0.9660

Appendix 2

Note on Independent Front Wheels, Parallelogram-Type—Broulhiet System

As illustrated in Figs. 32 to 37, the king pin of the steering knuckle is carried by a part (3) attached to the chassis at the upper end by a short lever (6) and at the lower end by two levers (7) and (9). The axes of all joints, on the same side of the chassis, are parallel in a slanting direction converging toward the front of the chassis. The braking couple of a front wheel is transmitted by the axis (10) which has the same dimensions as the king pin of the steering knuckle and the same bearings; all the other joints are mounted on Repusseau rubber Silentblocs.

Suspension is through a transverse spring the ends of which are joined with the lower lever (7) by a simple coupling; this spring consists of an arithmetic spacing of the leaves with rollers, that suppress friction, at the ends of the different leaves.

The tie-rod of the steering knuckles is broken at the center; these two halves are attached to a central lever (35) carrying three ball-joints, the third ball-joint receiving the steering drag-link. The axis of the central lever (35) is joined to the chassis; steering is therefore suspended and is not influenced at all by displacements of the wheels. The position of the three ball-joints is chosen in such a way as to overcome all of the variation in toe-in of the wheels during their entire displacement.

The working out of the parallelogram has been carried out according to the plan conceived by Julien and gives, as results, a variation in the parallelogram of the wheels of 0 deg. 55 min.; a variation of the track of 2 mm.; and the variation of the wheelbase is negligible.

As for the wheel displacements, with relation to the normal position upon the road, they are 80 mm. on compression of the spring and 40 mm. on expansion of the spring.

Appendix 3

Note on Independent Front Wheels of the Slipper Type—Broulhiet System

The transverse member at the front of the frame constitutes the front axle of the vehicle. See Figs. 38 and 39. For this purpose, this transverse member is bent in a horizontal plane at each end. These ends carry a conical groove in which is encased the king pin that we call "chandelle" (candle).

The steering-knuckle spindle is joined to a tubular piece that we call "cheminée" (chimney). This "chimney" pivots and slides on the "candle" over two ball bearings.

Suspension is through a transverse spring fixed at the center of the forward cross-member of the frame; owing to the bent form of this forward cross-member, this transverse spring can be placed in a vertical plane passing through the "candles."

The ends of the transverse spring are joined to the "chimneys" by inclined double shackles attached to a ring thrust-bearing turning on the "chimney."

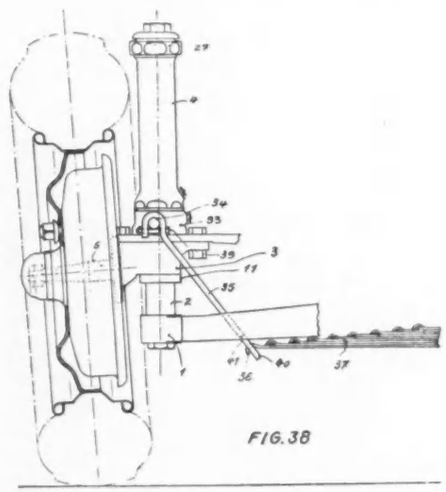


FIG. 38

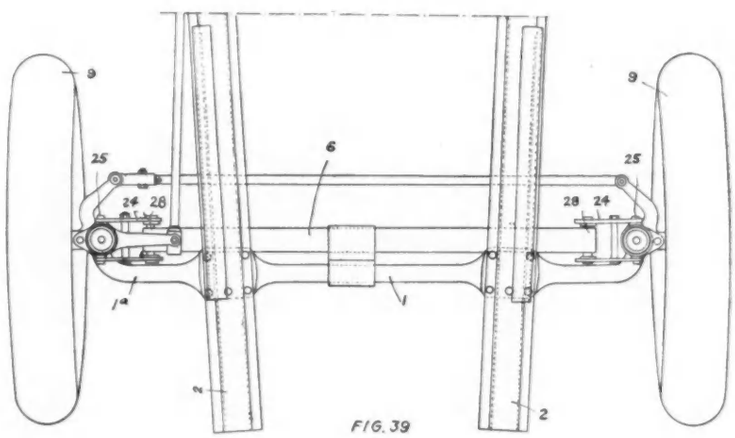


FIG. 39

Figs. 38 and 39—Broulhiet Front-Wheel System

The transverse spring has an arithmetic spacing of the leaves, with rollers at the ends of the different leaves for suppressing friction.

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Discussion of Broulhiet Paper

Suspension Types Will Be Developed for Each Country

—Tore Franzen
Experimental Engineer, Chrysler Corp.

IT seems as though every country has its own automobile problems, particularly as to suspensions, and it is apparent that when the country is big enough and the industry of that country is sufficiently alert, types suiting the particular country will be developed. An international all-purpose automobile is not yet here.

It is almost certain that a great deal of the tire wear is due not only to the spectacular type of driving that M. Broulhiet refers to but to poor alignment as well. The highly developed art of wheel alignment, as we know it in America today, is strange to the greater portion of the continental and English industries. The instrumentation we are used to seeing in better garages for truing-up front axles and for wheel alignment is not common even in some of the finer laboratories of builders of fine automobiles in European countries.

Six years ago I asked a prominent engineering executive of a large German company how many miles his firm got on a set of tires. At that time he said the maximum he had record of was 20,000 km. (12,427 miles) but the average was much lower. A month ago I asked him the same question and he gave me the same answer.

Under the heading "Fourth Condition," in M. Broulhiet's paper, it is possible to misunderstand the meaning of, "The oscillations are induced chiefly by the constant factor of damping." This sentence undoubtedly could be clarified.

M. Broulhiet does not touch upon the so-called "pendulum axle" as used in the designs of Tatra, Austra-Daimler, Daimler-Benz and others. This suspension has at least an academic importance and it would be interesting to have M. Broulhiet's comments on this.

A complete elimination of friction and building up of damping in sufficiently large shock-absorber units is unquestionably a great target to shoot at. It seems possible to reach very agreeable results without going the whole way by using leaf springs in which the friction is cut low and kept constant, as in the case of some springs used by our own organization where the coefficient is close to 0.17 per cent instead of the coefficient referred to in the paper, or 5 per cent.

In his reference to a German company that uses coiled springs on only one of its models, Mr. Litle is in error if he refers to the Daimler-Benz Co. as this company uses coiled springs on all its cars in all the rear suspensions. In the front supensions it uses coiled springs on its largest car; on the medium-size car, two coils and one leaf spring; and on the two small cars, two leaf springs.

It is thinkable that a suspension with a single arm in front of the axle can be constructed so that the dip due to brake action is not objectionable. This, however, does not necessarily include the complete usage of Dubonnet's design.

Satisfactory Units Depend Upon Engineering Skill

—J. H. Hunt
Director, New Devices Section, General Motors Corp.

THE Society is greatly indebted to M. Broulhiet for his clear exposition of many of the problems involved in installing independent-front-wheel suspensions on automobiles. His paper is well worth the careful study of anyone interested in this problem. It brings forward once more the fact that our engineering work must be a compromise between a large number of requirements, some of these being conflicting.

M. Broulhiet has emphasized the need for a rigid frame, and further has urged that the moment of inertia of the front end of the vehicle be made as great as possible. In a paper entitled Frame Design and Front-End Stability[1], Clyde R. Paton pointed out that the introduction of rubber suspensions

[1] See S.A.E. JOURNAL, August, 1932, p. 305.

had not only reduced the rigidity of the frame at the front but had also reduced the effective moment of inertia of the front end. M. Broulhiet has pointed out the need for eliminating, so far as possible, engine disturbances. The most effective way to accomplish this is by installing a flexible engine mounting.

At this session we have an example of conflicting advice. Alex Taub's paper on Engine Mounting will tell us how to secure a flexible engine mounting and, in so doing, will advocate reducing the moment of inertia which is effective to reduce frame twisting. At the same time M. Broulhiet is urging us to keep the moment of inertia up to a higher value. The easiest way to accomplish this would be to go back to a rigid engine mounting, which we know would be commercially impossible at present. Another example is that Sir Dennistoun Burney was urging us recently to mount the engine at the rear. Obviously, if this is done, the moment of inertia at the front end is again reduced. The engineer naturally will have to use great skill in selecting from these various prescriptions the particular expedients which will result in a satisfactory product.

Double-Spring Parallelogram Independently Sprung Design

—Wallace T. Miller
Lundelius & Eccleston Motors Corp.

M. BROULHIET'S paper is very interesting, even though very technical and somewhat theoretical. It rather leaves me in a haze, since it discusses many and various designs of independent-wheel suspension, whereas my mind has functioned on but one design, that of myself and my associates, at Los Angeles, Calif.

Nearly seven years ago I joined a group in Los Angeles to develop from its fundamental design an independently sprung job which we thought would fit the American needs as well as fitting American production methods. The design, as most of our engineers know, is a double-spring parallelogram. All torque from both engine and braking is taken through the springs without the use of any torque tubes or radius rods.

The car to which I refer is nearly six years old. It has more than 195,000 miles to its credit in eleven trips between Los Angeles and Detroit, plus tests upon nearly all mountain roads in the West, under all climatic conditions. In addition, it has had several speed tests made on the Indianapolis Speedway. I can confidently state that our experimental work is behind us, in this design.

In developing this car we had four cardinal points in mind: (*a*) it must not cost more than the equivalent-wheelbase car of conventional design in a comparable production schedule; (*b*) it must not upset the vogue of American style; (*c*) it must not present sales resistance; and (*d*) it must be much easier to service in the ultimate owners' hands. Most engineers who have analyzed our design by driving and riding it scores of thousands of miles have agreed that we have accomplished the cardinal points mentioned.

I can visualize at least fifty ways to spring a vehicle independently, but only one way exists to steer such a vehicle perfectly; it is to steer each front wheel individually from an absolute center, so that the shocks or oscillations of one wheel have absolutely no effect upon the other wheel and that the shocks or oscillations of either or both front wheels do not come into the steering wheels. I have such a steering gear on my car.

Suspension Problem Not Solved Merely by Independent Wheels

—Andre Dubonnet

FIRST, it must be remarked that independent-wheel springing is not a cure-all for the suspension problem merely because the wheels are independent of each other. This method of flexibly connecting wheels and chassis frame largely reduces the many difficulties encountered in the development of riding quality. With this system, much softer springs than are generally used will operate satisfactorily and without the usual disadvantages. There are many ways of mounting independently sprung wheels and it would be fortuitous that all be equally good.

To obtain riding comfort, steady steering and satisfactory road-holding ability, it is not sufficient to substitute the new principle for the old. The new arrangement can be applied more easily, which accounts for the large number of car manufacturers who are experimenting along that line at present.

It should be observed that, with an axle, when a wheel rides over an obstacle, its end of the axle moves up while the other remains at approximately the same distance from the ground; the axle is inclined relatively to the frame. As a result of the inertia of the frame, at least one wheel side-slips. Any spring deflection causes side-slippage between the tire and the road surface, with a resultant side-thrust against the frame. These side-thrusts, unavoidable with rigid axles, interfere with road-holding ability, and, at high speed over a rough road, the chassis cannot strictly follow its straight course.

To avoid transverse wheel-displacement, which interferes with road-holding ability, each wheel must act independently in its vertical motion. Suppression of transverse displacements is to the advantage of independent wheels. The wheel must never lose contact with the ground. Passing over a hole, it must drop freely and instantly with the road surface, which obtains when the ratio of sprung to unsprung weight is high. Lightness of the unsprung assembly also prevents it from being projected off the ground after coming over a bump.

Springing must be flexible and responsive to prevent the body's being subjected to small road-surface irregularities, which cause the car to shake transversely and to have poor road-holding ability. Load distribution is disturbed, with consequent variations in the adherence of the wheels to the road and in the reliability of their course. These transverse body-movements cause the springs to twist; but, if they are very flexible and undamped, the displacement will be greater. Hence, great flexibility, high sensitiveness and proper damping at the right time are required to prevent important reactions against the frame.

Front-end springing has a dominating influence on passenger riding comfort. First receiving road shocks, it disturbs load distribution and stability. As steering is through the front, anything interfering with stability must be avoided; such as wabble, tramp, shimmy, kick-back and gallop. For these reasons, steering parts must be sprung and independent of the vertical-suspension motion.

I define road-holding ability as the stability of a car along its course. Working out front-wheel independence, front wheels must move parallel to themselves, a condition necessitated by the fact that they have two functional movements; steering action about a vertical pivot and vertical-suspension motion.

The arrangement consists of a wheel-supporting arm articulated from a suspension box which itself pivots on the frame. This arrangement eliminates shackles and friction points, which would impair the sensitiveness of the suspension. It gives a simple system of sprung steering independent of the suspension. It offers a maximum of safety without shimmy and it allows large spring deflections (3½ in.) and flexibility (35 to 40 per cent) in front. These two advantages, however, can be obtained only with a spring suspension having variable flexibility, either inherently or through some special shock-absorber action.

The characteristics of the Dubonnet suspension are:

(1) Great flexibility in the fully loaded range

(2) Undamped free center for small oscillations (1½ in.)

(3 Variable flexibility to assure that, nothwithstanding recurring actuating causes of oscillation, the deflections will be kept within the range of the suspension

(4) Rebound damping

(5) Sensitiveness due to the absence of shackle or interleaf friction.

American Trend Seen Ignoring European Ideas

—T. J. Litle, Jr.
Engineer and Industrialist

THE paper impresses me with the fact that the real reason American engineers have not accepted any of the European independently sprung designs is because they have not as yet actually been perfected and are not practical for our mass production. M. Broulhiet has indicated clearly some of these shortcomings. The mere fact that there is no real trend toward similar design proves this, as does the violent disagreement between the tire manufacturers and car makers as to the necessity and importance of using tires balanced to the extreme limit. The same situation exists with regard to steering; our experience in this country bears this out.

To me, such references spell imperfect design resulting from hectic attempts of many inventors to cure troubles; and from a clear attempt to place the blame other than where it belongs. Entirely too much tinkering has been done and too many original designs have been abortive. M. Broulhiet's sudden conclusions condemning leaf springs are most surprising. Some European makers are using many groups of small coiled springs for the simple reason that they had to compromise their designs to avoid excessive spring length and range of action in the case of a single spring, especially on the rear-end suspension. Where large groups of smaller springs are used they are augmented by a multitude of guides, piston links, check valves and other gadgets totaling hundreds of parts that require servicing.

The paper correctly states that the Germans do not favor coiled springs. I know of only one maker who uses them, and then only on his lightest car; on his larger cars he uses leaf springs. It is a positive fact that the greatest trouble abroad lies in steering geometry. My own observations in America indicate a trend toward original design almost completely ignoring European traditions.

I drove from Detroit to Chicago in 5½ hr. in a car built and designed in California. It has been road-tested nearly 200,000 miles. The steering is remarkably easy and eminently safe. On one test a tire was blown out at 70 m.p.h. with only a 3-in. shift at the steering wheel, and the car was still controllable with only slight effort. To date, 20 tires have been used. The present set probably will continue to 220,000 miles. Three different makes of tires were on test, all being picked from regular stock. An average tire life of over 40,000 miles clearly indicates that a properly designed independently sprung job is far easier on tires.

Two different makes of standard steering gears were tested with equally perfect success, wear being unobservable. To refute French claims that tires have to be balanced with extraordinary precision, a tire out of balance 60 in-oz. was successfully used. My own preference is for the double-spring-parallelogram mounting as representing the ultimate in simplicity and the cheapest possible construction. On the trip from Detroit I read aloud with perfect ease in the tonneau at 72 m.p.h.

While not agreeing with M. Broulhiet on many points, I thoroughly sympathize with him in his futile efforts to obtain cooperation from some of the French engineering organizations because I have been on both sides of the fence. I might add that an engineer is not without honor save in his own company.

Study of Other Factors Brings Different Results

—H. O. Fuchs

M. BROULHIET'S results are convincing, not only due to the weight of his experience but also to the lucid manner in which he deducts them theoretically. On one point, however, a different result is reached by taking into consideration other factors.

The adherence of the wheel to the ground is diminished by shock absorbers which check the extension (rebound) of the spring. As M. Broulhiet has pointed out, this adherence is very important, and the better adherence to the ground is one of the advantages of independent springing. This is due to the reduction of unsprung weight. The adherence to the ground depends upon the speed with which the wheel returns to the ground after jumping over an obstacle. This speed is proportionate to the ratio of unsprung weight to sprung weight. As the force exerted upon the unsprung mass by the spring is substantially equal to the unsprung weight, the acceleration of the wheel toward the ground is:

$$a = g\left(\frac{W\ \text{Sprung}}{W\ \text{Unsprung}} + 1\right)$$

where g is the acceleration due to gravity and W is the weight. By introducing a shock absorber, this movement is made slower.

For the same shock absorber, the decrease in the speed of the wheel toward the ground is greater with small unsprung-weight than with great unsprung-weight. This decrease in

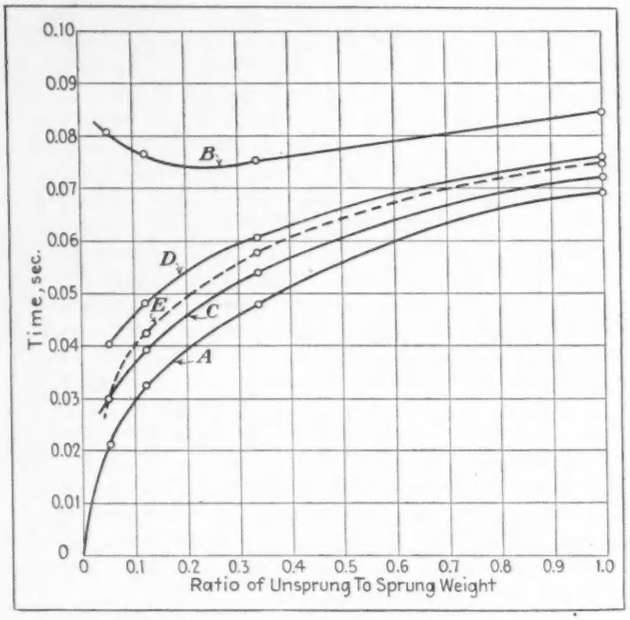

Fig. 1—Time Which the Wheel Spends in the Air before Retouching the Ground

Curve *A*, without damping; Curve *B*, with aperiodic critical damping; Curve *C*, with one-quarter critical damping; Curve *D*, with one-half critical damping; and Curve *E*, with damping similar to that for Curve *D*, but limited to a maximum force of 330 lb.

speed is so important that, for the same car and shock absorber, the movement can become slower with small unsprung-weight than with great unsprung-weight, and thereby all the advantages of low unsprung-weight can be lost.

The time which the wheel spends in the air before retouching the ground after a jump 2 in. high has been calculated for different ratios of unsprung to sprung weight and for different shock absorbers. In Fig. 1, this time is shown graphically against the ratio of unsprung to sprung weight.

Curve *A* in Fig. 1 shows time without using a shock absorber. Curve *B* is with a shock absorber as described by M. Broulhiet, with aperiodic critical damping and without limitation of the force to a certain maximum. The detrimental effect of the shock absorber is obvious. Curve *C* shows the effect of damping with one-quarter critical value and without limitation of the force. Curve *D* shows the effect of damping with one-half critical value; and Curve *E*, with damping similar to that for Curve *D*, but limited to a maximum force of 330 lb. Curves *A* to *E* show the advantage of such a limitation, which easily can be obtained by a relief valve or a valve with relief openings, as shown by M. Broulhiet for the compressions (in-stroke). For cars with light unsprung-weight, this point becomes even more important than it is for our present standard cars.

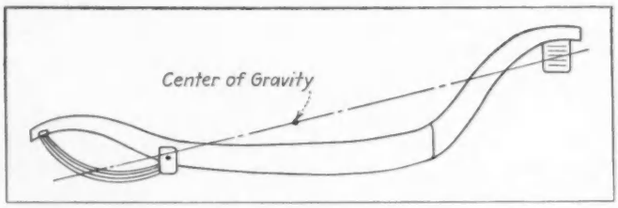

Fig. 2—Diagram Illustrating how Moments Due to Centrifugal Forces and Curves Are Avoided by Locating the Plane of Suspension so that It Passes Through the Center of Gravity

Regarding another point in M. Broulhiet's paper, an additional remark may be interesting. Complete equilibrium in the curves is obtained by a German manufacturer. He places the plane of suspension so that it passes through the center of gravity. In this way he avoids all moments due to centrifugal forces and curves, as shown in Fig. 2.

Consideration of Practical Applications of Damping

—Georges Broulhiet

THE disadvantage of damping which Dr. H. O. Fuchs mentions, that the wheel is braked in its movement toward the ground, exists. I have tried to solve these questions by calculations and experiments, but they would require too much space to reproduce here in full. Therefore, I will give only a short survey, which shows the points which are valuable in practical applications.

(1) The braking of the return of the wheel to the ground would be very disadvantageous in the presence of successive oscillations. However, a good shock absorber reduces the movement of the springs to one-half cycle and the loss of adherence to the ground during this half cycle of oscillation is negligible, as it occurs very seldom at the same time for both wheels in a car equipped with shock absorbers.

(2) The shock absorber which is part of my design for a front axle with vertical guide *(chandelle)* works in a manner which I did not describe; it works only when the wheel is below the equilibrium position with regard to the car. In this way, the objection raised by Dr. Fuchs is overcome. There is another reason for the use of such a shock absorber. It is that on an axle with vertical guides, the movement downwards from the equilibrium position is smaller than 45 mm. (1¾ in.) on roads. By this special shock absorber, I have limited the movement of the axle to 45 mm. below the equilibrium position.

(3) One can choose between two kinds of shock absorbers. First, those with a resistance varying in proportion with the velocity of the movement and without regard to the position. These instruments are not affected in their functions by the weight of the freight or passengers. Or, second, those with a resistance which depends on the position of the axle; they function to brake the movement only below the equilibrium position.

I prefer the first kind for ordinary axles and flexible springs, because they are easier to build and because these springs will be influenced very much by the loading of the car. I make use of the second kind when the movement of the axle below the equilibrium position should be limited; they avoid, at the same time, the disadvantage which Dr. Fuchs pointed out. Both solutions give practically the same results. This proves that the performance of shock absorbers and the manner in which their resistance is influenced by velocity and position may be adjusted in different ways, giving the same result as long as the energy destroyed by the shock absorber, the negative work done by it, remains the same. This is the main condition.

How to Relate Body Tooling Budgets to Quantity Requirements

By N. H. Manning
Briggs Manufacturing Co.

HAVING pointed out that one of the industry's most serious errors is overestimating its requirements, Mr. Manning shows how body tool budgets can be set up in relation to the quantities expected to be produced.

While shunning the argument regarding relative merits from a design standpoint of the all-steel and the composition body, he does approach the question from a purely manufacturing standpoint when he says:

"It will be found that certain designs lend themselves more readily to steel construction and, from a cost standpoint only, wood may be eliminated to advantage. It will become apparent, therefore, that at some point in the estimated production quantities you will find from a purely economical standpoint wood being replaced by formed steel sections in whole or in part."

IT is not the purpose of this paper to attempt to analyze or solve anyone's particular problem, but rather, having due regard for the variables involved, to deal with the subject in a general way, with a few specific references which may prove of interest.

The present day conventional five-passenger sedan may be used as a basis for comparison with other supplementary models. Let us assume that this body has the customary sloping front end, cowl ventilators, metal above and below the belt, and let us analyze the problem as to how this body can most economically be produced for a given quantity production.

As the prices of labor and material are rapidly changing and subject to further change, any figures used must be taken as approximate and used for comparative purposes only.

If one body only is required, the engineering department would supply a paper draft consisting of an outline of the body, with a few of the more difficult sections in detail. The wood and metal parts would for the most part be made by hand, with the aid of a few simple forms for shaping the more difficult metal parts. Wood patterns would be used for all castings and for any parts where castings could be substituted for sheet metal parts. The cost of such a job would probably run not less than $4000, depending of course on intricacies of design and construction.

Should we desire to duplicate this job, producing, let us say, 25, our procedure will alter somewhat. Our paper draft will still suffice. On account of the increased quantity the wood parts can be more economically produced with the assistance of a few flat patterns and shaper forms for the more difficult wood shapes. Beyond the duplication of parts the operations would still have to be produced for the most part by hand. The cost for making such patterns, forms, etc., should not exceed $2500 to $3500.

If the quantity be increased to 25 or more, and considered in runs of 50 or 100 up to about 200 units, it will be found that there is a very definite point at which the savings in labor will more than compensate for the expense of a simplified form of tooling. This breaking point usually occurs at between 20 and 30 jobs, depending of course on the design and construction. This simplified form of engineering and tooling would consist of the following:

1. An aluminum draft for accuracy in making layouts and shaper forms.
2. Wood layouts and shaper forms would be made for producing the wood parts, with a view to eliminating excessive hand-labor.
3. Artz press forms or the equivalent would be used for producing cowls and similar metal parts. Hand hammer forms and a few simple dies would be made for producing the balance of the metal parts.

The expense of the above should not exceed $9500 and should produce economically up to 100 units. With a few additions to metal forms and dies, and with an increase in expenditure up to, say, $12,000 or $15,000 we could economically produce up to 200 units.

Assuming now that the quantity be increased from 200 to 500 units, we reach a certain high-grade car market, which is common to many manufacturers and which for competitive reasons demands that labor costs be reduced very considerably. To accomplish this we would probably modify our engineering and tools as follows:

1. Draft and details would be enlarged and completed so that parts could be lined up for production at more nearly production prices.
2. Layouts and shaper forms would be improved so that all shapes could be produced without further hand labor and otherwise manufactured in the most economical manner.
3. Artz press forms could still be used for cowls and other similar shapes. Hand hammer forms would give place to inexpensive forming dies, the reflanging and trimming operations still remaining to be done by hand.

Excerpts from paper presented at S.A.E. International Automotive Engineering Congress, Chicago, Aug. 31, 1933.

4. Welding forms and assembly jigs would also be improved to facilitate assembly.

The expense of this may run from $15,000 to $25,000, depending on the construction and methods of manufacture adopted.

From 500 to 1000 we would find that our engineering and tools could be increased to $30,000 or more, which increased expense would be more than offset by economies in labor. This improvement in tooling would be brought about principally through improvements and additions to dies, welding forms and assembly jigs.

Tooling for Large Quantities

Our next quantity unit to be considered would be the 1000 to 3000 range, which is probably the most difficult quantity for which to provide an economical tooling, as on this will depend the type and quality of body to be built. If the unit is to be used on a car priced above $2000, we would probably be justified in spending up to $200,000, or more, especially if some of the parts could be made interchangeable with other models; and, considering that our body is of the more expensive type, the unit tool cost per body would not be considered excessive.

If on the other hand we have to tool for a body to be used on a car within the cheaper bracket, we would in all probability keep this cost as low as possible, in which case it might not exceed $60,000 to $75,000.

The difference in the tool line-up between this and the previous quantity would be briefly as follows:

1. Our draft and detail would be completed to the point where all parts could be produced by production methods and at production prices.

2. Shaper forms and assembly jigs for producing and assembling wood parts would be improved to the point where all hand labor would be eliminated.

3. All metal parts would now be made as die stampings, and trimming and reflanging dies would be introduced in a limited way only.

4. Assembly and welding fixtures would also be improved to reduce hand labor as much as possible.

Steel Construction Discussed

Up to this point we have assumed that our bodies have been of composite wood and steel construction. There is a point, however, where the combination of wood and steel must be considered in contrast with the all-steel construction. It is not the purpose of this paper to discuss the merits or disadvantages of either method of construction. It will, however, be found that certain designs lend themselves more readily to steel construction, and, from a cost standpoint only, wood may be eliminated to advantage. It will become apparent, therefore, that at some point in the production you will find, purely from an economical standpoint, wood being replaced by formed steel sections in whole or in part.

In considering, therefore, quantities of, say, 3000 to 12,000, we shall find steel front ends and all steel doors being introduced and economies effected, assuming of course that we favor such construction. The cost of such tooling may run as high as $300,000 to $350,000.

Owing to the larger quantity of units involved, the field becomes more highly competitive, so that our tooling must be modified to produce such units at the lowest possible labor costs consistent with tooling for semi-production quantities.

We now arrive at our last and most competitive range, namely, what we may term continuous mass production, which necessitates providing for tools which are to be used continuously for a period of a year or more, for the production of as many units as may be required.

If we adopt an all-steel construction, we may be justified in spending up to $450,000 or $500,000. If composite construction, the cost will be somewhat less, probably 25 per cent.

It would be impossible at this time to go into all the details of tooling for a production model, and so we will confine ourselves to a few general remarks in this connection.

This field produces by far the greater volume of our requirements, and in many cases provides the earning power on which the balance of the industry survives. Owing to the extremely competitive nature of the product, not only must tooling be provided to reduce labor costs to a minimum, but operations must also be combined to provide maximum savings in material.

It is here, therefore, that the best engineering and production talent is called into action and improvements are daily being made to effect savings in both labor and material. Not only must essential operations necessary to the construction of the body receive attention, but also plant layout and handling of materials in proper sequence must be given due consideration because they play an important part in the reduction of labor.

Tools and Body Types

If we are concerned with a composite body, all wood parts will be reduced to a minimum, both in quantity and section, consistent with strength and ability to render satisfactory service in the field.

Dies for producing panels will be perfected to the point where all possible metal finish can be eliminated in obtaining a suitable paint surface. Dies for trimming, reflanging and clinching operations will be improved as much as possible to eliminate hand labor.

Assembly and welding jigs will also be improved to combine and eliminate all unnecessary operations.

If our body is of steel construction, the procedure will be in the direction of producing the same result, and will differ only in details of accomplishing these results, due to the difference of construction.

It would seem, therefore, that before undertaking any new program for tooling, the engineering department should make a careful survey of the entire situation, viewed from the standpoint of management, sales and manufacturing, and that from data so obtained, a decision should be made as to the classification into which the particular problem falls, which for lack of better terms we may describe as follows:

1—Custom—Quantity up to 25
2—Semi-Custom—Quantity 25, 100, and 200
3—Semi-Production—Quantity 200, 500, and 1000
4—Production—Quantity 1000, 3000, and 12,000
5—Continuous Production—Quantity 12,000 and up

For each classification it will then be possible to determine just what tool expense will be justified in arriving at a minimum total unit body and tool cost.

TYDOL was the fuel, Veedol the oil used in the winning car at the Indianapolis Race. In its article about the race in the July issue, the S.A.E. JOURNAL erroneously stated that the oil used was Tydol.